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Third Progress Report

SPECIAL COMMITTEE ON STRESSES IN RAILROAD TRACK

1922

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Third Progess Report

OF THE

Special Committee on Stresses in Railroad Track

1922

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To the American Railway Association:

The Special Committee on Stresses in Railroad Track herewith presents its third progress report.

I. INTRODUCTION

1. **Preliminary.**—Since its organization in 1914 the Committee has been coöperating with the Special Committee to Report on Stresses in Railroad Track appointed by the American Society of Civil Engineers, the membership of the two committees being the same with the exception of one member of the American Railway Engineering Association who does not hold membership in the American Society of Civil Engineers, and the work has been carried on as one committee. This report is presented simultaneously to the two societies and to the American Railway Association, which also has been coöperating in the work for the past four years.

As stated in the earlier reports, the Committee has felt that an adequate report on stresses in railroad track must be based largely on experimental data derived from extensive tests on standard railroad track, and that in view of the complexity of the action of track under load and the variability of the condition to be found in track and load, the work of conducting experiments and reducing the data would necessarily require a large amount of time and effort. The development of methods of conducting the tests and the work of devising the instruments and apparatus has involved a considerable expenditure of time, effort and money. It has been recognized that in obtaining data on the action of track under variable conditions of both track and load great refinement of method was not possible and that it was important to make tests under conditions

of railroad service as nearly normal as is possible, also utilizing of course the data of laboratory investigations where conditions would not warrant satisfactory experiments in the field.

The first progress report was published in Vol. 19 of the Proceedings of the American Railway Engineering Association (1918). It may also be found in Vol. 82 of the Transactions of the American Society of Civil Engineers. In this report a method of analysis for determining the moments and stresses in the rail by considering the track as an elastic structure was presented, the instruments and methods of testing were described, the depression of the track and the rail profile under load were given, and the stresses developed in the rail under a variety of loads were recorded. The report relates principally to stresses in rail and to the general elastic conditions of the track.

The second progress report of the Committee was published in Vol. 21 of the Proceedings and in Bulletin 224. It may also be found in Vol. 83 of the Transactions of the American Society of Civil Engineers and in Circular No. S-II-10 of the American Railway Association. This report gives the results of tests on two railroads with several types of locomotives to determine the stresses in rail in relation to speed and counterbalance effect. It records the results of tests on two railroads to determine the depression of track and the flexure of ties and their action under load for a variety of conditions found in track. Tests to find the manner and the principles involved in the transmission of pressure from one or more ties downward and laterally through ballast material were reported and an analytical consideration of the transmission of pressure given.

The work herein reported includes tests on straight track and curved track. The tests on straight track considered the effect of counterbalance, the effect of speed and the combined effect of speed and counterbalance, and the lateral bending moments and stresses in the rail. For the curved track there are taken up the general action of curved track as the locomotive traverses the curve, the magnitude of the vertical bending stresses in the rail and the corresponding vertical loads producing them as contrasted with the normal loads, the lateral bending moments and stresses under the several wheels, the distortion of the alinement of the curves, and the general effect of speed, degree of curve and superelevation. The tests were conducted on the Illinois Central Railroad, the Delaware, Lackawanna & Western Railroad, the Atchison, Topeka & Santa Fe Railway, and the Southern Pacific Company. Eight distinct types of locomotives were used; in several types there were variations in the design of the locomotives of the different railroads and the number of locomotives used in the tests was thirteen. The report is presented under the heads of Tests on Straight Track and Tests on Curved Track.

The Committee is continuing work on the subject assigned to it.

2. **Acknowledgment.**—Since the second progress report was issued the funds for use in carrying on the work of the Committee have

been taken principally from contributions made by the American Railway Association. The Committee expresses its appreciation of this support. Expenditures have also been made from funds in the hands of the American Society of Civil Engineers and the American Railway Engineering Association contributed by the United States Steel Corporation, the Bethlehem Steel Company, the Lackawanna Steel Company, and the Cambria Steel Company.

The coöperation of railroad companies in furnishing facilities for the test work has itself been a large contribution. The Illinois Central Railroad, A. S. Baldwin, Vice-President, and F. L. Thompson, Chief Engineer; the Delaware, Lackawanna & Western Railroad, G. J. Ray, Chief Engineer, and A. J. Neafie, Principal Assistant Engineer; the Atchison, Topeka & Santa Fe Railway, C. F. W. Felt, Chief Engineer, have been liberal in providing locomotives, track, and other facilities in the conduct of the tests. The Atchison, Topeka & Santa Fe Railway and the Southern Pacific Company have also made it possible for the Committee to use such portion of the results of the tests made on the lines in California as are related to the subject-matter of this report. A number of engineers from the engineering, mechanical, and test departments of the Atchison, Topeka & Santa Fe Railway gave helpful assistance throughout the work. The services of Mr. C. C. Huycke in aiding in the reduction of the data were also contributed.

Mr. E. E. Cress, Assistant Engineer of Tests in charge of field and office work and the reduction of data and preparation of material, is entitled to much credit; his thorough familiarity with details, keen grasp of the problem, and helpfulness in the study and preparation of the data have made his services particularly valuable. Mr. Louis J. Larson, Associate in Theoretical and Applied Mechanics in the University of Illinois, by reason of his special training and insight in testing work, has given valuable service in both laboratory and field work during the summer seasons. H. L. Parr and R. Ferguson have given helpful service in the reduction of data and the preparation of the report. Others have assisted in the work from time to time and all have given loyal and careful service.

The University of Illinois has continued to coöperate in the work by giving the use of laboratory, shop, and office facilities, and through the service of members of the staff of the Engineering Experiment Station from time to time.

The Committee records with profound regret the loss of a member by death, Mr. Archibald Stuart Baldwin. Mr. Baldwin from the time of the organization of the Committee took a great interest in the work and was active in the making of plans, the provision of means for carrying on the tests, and the discussion and interpretation of the results. He was especially helpful in making the arrangements for the extended use of locomotives and track of the Illinois Central Railroad for test work.

3. **The Track.**—The test on the Atchison, Topeka & Santa Fe Railway, made in September and October, 1920, were conducted on a

section of single track of the 10° curve at the west approach to the bridge over the Mississippi River at Fort Madison, Iowa, and on the eastbound tangent track in Illinois some two miles east of this point; also on tangent track and curves (single track) near Ribera, New Mexico, about twenty-seven miles west of Las Vegas. The location, weight of rail and its date of laying, tie spacing, depth of ballast, grade, gage, superelevation of outer rail, and corresponding speed are given in Table 2. The ties were 6 by 8 in. by 8 ft., mostly oak in the test tracks in New Mexico and Iowa, and pine in Illinois. Tie plates were in use on all ties. No curve braces were in use. The ballast in the track in New Mexico was rock and that in Iowa and Illinois was gravel. The test section of the 10° curve at Ribera was on a fill of 12 to 14 ft.; the 10° curve at Fort Madison was on 25-ft. fill. The 6° curve at Ribera was in a shallow cut and the tangent on ground nearly at grade. The tangent in Illinois was on 6-ft. fill. The rails were laid with alternate joints. The track was used as it was found. The 10° curve at Fort Madison had recently been resurfaced and realined and was in excellent condition. The other track had not been tamped recently, but it was in fair surface and alinement.

The tests on the Delaware, Lackawanna and Western Railroad, made in July and August, 1920, were conducted on eastbound tangent track at Dover, New Jersey, on eastbound 4° curve east of Dover, on eastbound 6° curve east of Mt. Tabor, New Jersey, and on eastbound 7½° curve at Paterson, New Jersey. Information about the track is given in Table 2. The ties were 7 by 9 in. by 8 ft. 6 in.; creosoted oak ties, creosoted oak and pine ties, and creosoted pine ties, respectively, were used on the three test sections. Tie plates were used throughout. Screw spikes were in use; the 7½° curve also had some driven spikes. The ballast was rock, rock on cinder, rock on gravel, and rock on sand. The tangent was a light fill, the 4° curve nearly at grade, the 6° curve on 10-ft. fill and the 7½° curve in 5-ft. cut. The rails were laid with alternate joints. The track was tested as found and was in good surface and alinement.

The tests on the Illinois Central Railroad made in June and July, 1920, were conducted on a 7° curve at Murphysboro, Illinois, and on a 14° curve at Champaign, Illinois. The track was tested as found; the surface and alinement were in fairly good condition. The 7° curve had rock screenings for ballast. The 14° curve had cinder ballast. The gage of the 7° curve was 4 ft. 9⅛ in.; that of the 14° curve 4 ft. 8⅞ in. Tie plates were in use.

The track at Bealville and Cajon, California, used in tests in 1922, is described in the discussion of the tests at these two locations.

It is seen from Table 2 that there are some differences in the gage of the track. For example, on straight track the gage on the test section of the A. T. & S. F. Ry. at Ribera was 4 ft. 8½ in., at East Fort Madison 4 ft. 8⅝ in., and on the D. L. & W. R. R. at Dover 4 ft. 8¾ in. The gage of the 6° curve of the D. L. & W. R. R. at Mount Tabor was 4 ft.

8¾ in., and that of the A. T. & S. F. Ry. at Ribera 4 ft. 8⅞ in. The gage of the 10° curve at Fort Madison was 4 ft. 9⅛ in., at Bealville 4 ft. 8⅞ in., while the track at Cajon was spiked to standard gage of 4 ft. 8½ in. at the time of the test. The foregoing values were obtained by averaging several measurements. A variation of as much as ¼ inch may be found in a distance of 5 or 10 feet. It is to be expected that the gage will widen with use after realinement, though measurements made at given point showed no change after four or five days of testing work.

The recommendation of the American Railway Engineering Association, as given in the Manual, is to use standard gage (4 ft. 8½ in.) for straight track and for curves up to and including 8°. It is also recommended that gage be widened one-eighth inch for each two degrees or fraction thereof over 8° to a maximum of 4 ft. 9¼ in. It is stated that

TABLE 1

Properties of Sections of Rails Used in the Tests

Rail Section (full-section)	Area in Square Inches	Moment of Inertia		Section Modulus		
		For Horizontal Axis	For Vertical Axis	Horizontal Axis		Vertical Axis
				Base	Head	Base
85-lb. A. S. C. E.	8.3	30.1	.....	12.2	11.1	.....
90-lb. A. R. A.-A.	8.8	38.7	7.5	15.2	12.6	2.9
90-lb. S. F.						
92-lb. Frictionless	9.0	32.0	8.3	14.7	10.1	3.1
101-lb. D. L. & W.	9.9	38.9	9.5	15.8	13.1	3.5
105-lb. D. L. & W.	10.2	50.2	9.5	18.0	15.7	3.5

under ordinary conditions it is not necessary to regage track if the increase in gage has not amounted to more than one-half inch, providing such increase is uniform, but the gage including widening due to wear should never exceed 4 ft. 9½ in. The foregoing recommendation would make the gage of a 10° curve 4 ft. 8⅝ in., with an allowable increase of ½ inch before realinement.

4. The Locomotives.—In Fig. 1 and 2 are given the locomotive diagrams for the locomotives used on the tests of track on the A. T. & S. F. Ry. in New Mexico, Iowa, and Illinois, and in Fig. 3 for those used on track of the D. L. & W. R. R. in New Jersey; wheel loads and spacings, diameter of drivers, crank pin radius, and the number and type of the locomotive are noted. In Fig. 4 is given a diagrammatic representation of the equalizing system of six of the locomotives. Tables 3 to 7 give data of the counterbalancing of the drivers of the locomotives of the A. T. & S. F. Ry., as furnished by the Mechanical Department of that

road, and Tables 8 and 9 similar data furnished by the D. L. & W. R. R. The footnotes give the approximate modification in the counterbalance obtained by the method outlined in the second progress report for finding the effect of the outside rotating parts not being in the plane traversed by the counterweights.

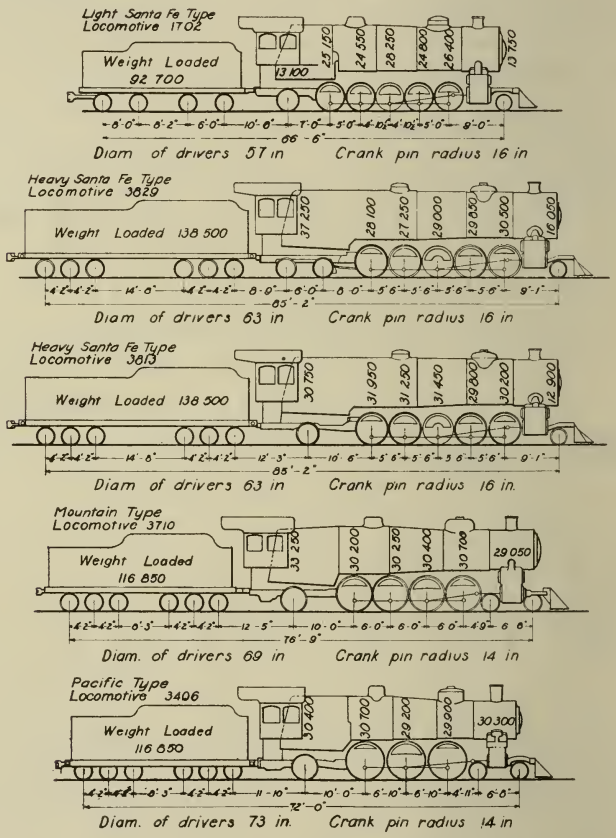


Fig. 1.—Diagrams of Locomotives of the A. T. & S. F. Ry.

The locomotives were used as they were found in service; all were in ordinary working order. Contours of the tires of the driving wheels were taken; they may be said to have been in average condition—neither newly turned nor badly worn. No measurement of the lateral play of the driving axles in their boxes was made. The practice of the A. T. & S. F. Ry. is to provide a total lateral play of 1/8 to 3/16 in. at the boxes of the drivers when the locomotive leaves the shop. The regulations of the Interstate Commerce Commission provide that this lateral play shall not be permitted to exceed ¾ in.

The Santa Fe type locomotive received the name from its introduction by the A. T. & S. F. Ry. The 1674 class, the first of this type to be built by any railroad, is here called the light Santa Fe type. No. 1702, the one used in the test, was built in 1903. The small size of the drivers

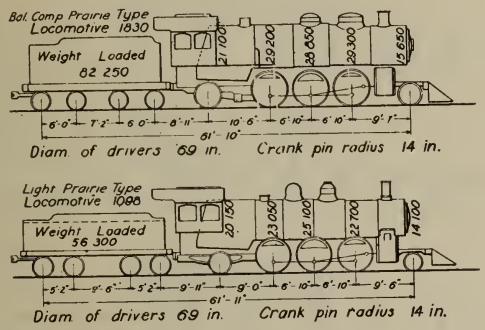


Fig. 2.—Diagrams of Locomotives of the A. T. & S. F. Ry.

made full counterbalancing on the main driver impracticable, and counterbalancing devices, known as bobs, were attached to the driver axle inside the bearings. As the driver loads and spacing of these locomotives are relatively small, the stresses developed in the rails used were smaller than those found with the other locomotives, even though the effect of speed, counterbalance and lateral bending was considerable. They were, of course, built for use on lighter rail sections. It should be noted that many

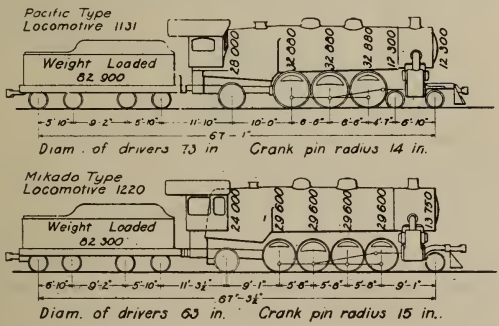


Fig. 3.—Diagrams of Locomotives of the D. L. & W. R. R.

Santa Fe type locomotives with 57-in. drivers still remain in use on a number of railroads of the country, and some of these are not of as good design as the 1674 class of this railroad. The Heavy Santa Fe type locomotive was not only much heavier but the design was greatly improved and

its general performance on the track was much better than the lighter locomotive; No. 3813, the one used, had been in service about a year, as had the Mountain type and the Pacific type of the A. T. & S. F. Ry. Those of the Prairie type had been in service fifteen to twenty years. The

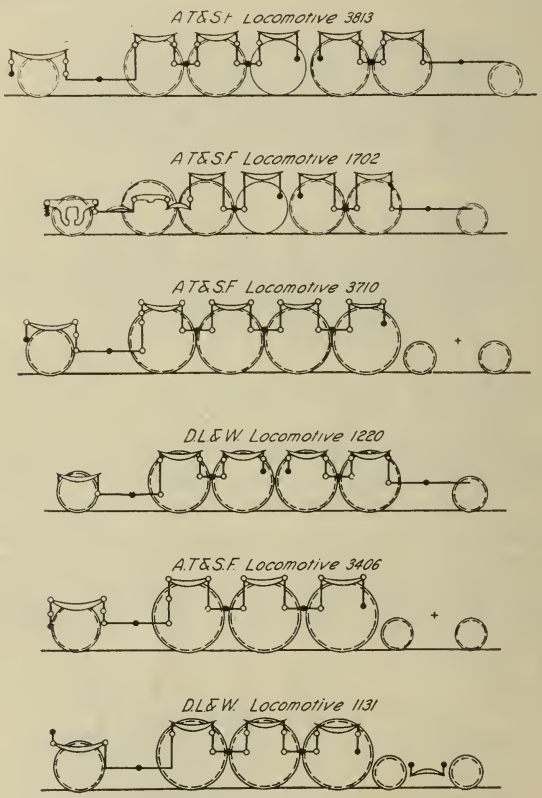


Fig. 4.—Diagrammatic Representation of Equalizing Systems of Locomotives Used in the Tests.

only change made in the locomotives during the tests was in the Mountain type; the lateral spring holding the trailer was removed for the last part of the tests. The Double Trailer Heavy Santa Fe type is the term given to the one locomotive which the railroad had equipped with four trailing wheels for experiment purposes; this is the only locomotive to which the double trailer had been applied.

The Mikado type locomotive of the D. L. & W. R. R. was built in 1913 and the Pacific type used was built in 1915. These two types of locomotives were also used in the tests made on this railroad in 1916.

TABLE 2

DATA OF THE TRACK

Tie Spacing, Depth of Ballast and Superelevation are Given in Inches and Gage in Feet and Inches

Track	Location	Rail Section	Date of Laying	Tie Spacing	Depth of Ballast	Grade per Cent	Gage	Superelevation of Outer Rail	Speed of Superelevation Miles per hour
Tangent.....	Ribera....N. Mex.	90-lb. S. F...	1917	18	12	−1.40	4-8½	....	.....
6° Curve....	Ribera....N. Mex.	90-lb. S. F...	1917	18	10	−1.18	4-8⅞	4.7	35
10° Curve....	Ribera....N. Mex.	90-lb. S. F...	1920	19	10	−1.02	4-8⅞	4.7	27
Tangent.....	E. Ft. Madison, Ill.	85-lb. A. S. C. E.	1906	20	22	0.0	4-8⅝	....	.....
10° Curve....	Ft. Madison....Ia.	90-lb. S. F...	1920	21	14	0.0	4-9⅛	2.0	17
Tangent.....	Dover, N. J.....	105-lb. D. L. & W.	1915	22	12	−0.55	4-8¾	....	.....
4° Curve.....	Dover, N. J.....	105-lb. D. L. & W.	1919	22	18	−0.63	4-8⅞	3.7	37
6° Curve.....	Mt. Tabor, N. J...	92-lb. Frictionless..	1914	22	6	−0.84	4-8¾	8.5	46
		101-lb. D. L. & W.	1915						
7½° Curve...	Paterson, N. J.....	105-lb. D. L. & W.	1919	20	9	0.0	4-9	6.4	36
10° Curve....	Bealville, Calif....	90-lb. A.R.A.-A.	1921	22	5 in. on old grade	2.00	4-8⅞	4.4	26
10° Curve....	Cajon, Calif.....	90-lb. S. F...	1920	18	10	1.80	4-8½	5.4	29
7° Curve....	Murphysboro, Ill..	90-lb. A.R.A.-A.	1914	20	12	0.0	4-9⅛	3.7	28
			1919						
14° Curve....	Champaign, Ill....	85-lb.....	1903	19	Cinder	0.0	4-8⅞	3.8	20

TABLE 3

Weights of Rotating and Reciprocating Parts, in Pounds—Pacific Type Locomotive of the A. T. & S. F. Ry.

RECIPROCATING PARTS

Piston	706
Crosshead	606
Union link and lower end of combination lever	60
40% main rod, weight on crosshead pin	433
	1805

$$\frac{.50 \times 1805}{3} = 301$$

ROTATING PARTS

Driver number	3	Main	1
50% of weight of reciprocating parts	301	301	301
Main rod, weight on crank pin	.....	650	.....
Side rod, weight on crank pin	190	705	190
Crank pin	117	367	117
Crank pin hub	89	153	89
One-half of eccentric crank	.....	82	.....
Total	697	2258	697
Equivalent weight required at center of counterweight	372	1435	372
Weight used	372	1435	372
Difference in weight required and weight obtained	0	0	0
Equivalent difference at crank pin center	0	0	0
Overbalance or underbalance at crank pin circle for rotating parts only	+301	+301*	+301

*If the effect of the rods and pin not being in the plane of the counterweight is calculated by the method given in the second progress report, this becomes approximately an underbalance of 35 lb.

TABLE 4

Weights of Rotating and Reciprocating Parts, in Pounds—Light Prairie Type Locomotive of the A. T. & S. F. Ry.

RECIPROCATING PARTS

Piston	752
Crosshead	392
Weight of main rod on crosshead pin	297
	1441

$$\frac{.666 \times 1441}{3} = 320$$

ROTATING PARTS

Driver Number	3	Main	1
Two-thirds of weight of reciprocating parts	320	320	320
Main rod, weight on crank pin	.....	297	.....
Side rod, weight on crank pin	163	1100	170
Crank pin	100	280	100
Crank pin hub	90	160	90
Total	673	2157	680
Equivalent weight required at center of counter weight	410	1630	414
Weight used	410	1630	414
Difference between weight required and weight obtained	0	0	0
Equivalent difference at crank pin circle	0	0	0
Overbalance or underbalance at crank pin circle for rotating parts only	+320	+320	+320

TABLE 5

Weights of Rotating and Reciprocating Parts, in Pounds—Mountain Type Locomotive of the A. T. & S. F. Ry.

RECIPROCATING PARTS

Piston	904
Crosshead	610
Union link and lower end of combination lever	62
40% main rod, weight on crosshead pin	522
	2098

$$\frac{.50 \times 2098}{4} = 262$$

ROTATING PARTS

Driver Number	4	3	Main	1
50% of weight of reciprocating parts	262	262	262	262
Main rod, weight on crank pin	.....	.....	783	.....
Side rod, weight on crank pin	180	530	665	175
Crank pin	135	130	430	100
Crank pin hub	129	129	266	129
One-half eccentric crank	.....	.....	87	.....
Total	706	1051	2493	666
Equivalent weight required at center of counterweight	417	640	1939	417
Weight used	417	640	1939	417
Difference between weight required and weight obtained	0	0	0	0
Equivalent difference at crank pin circle	0	0	0	0
Overbalance or underbalance at crank pin circle for rotating parts only	+262	+262	+262*	+262

*If the effect of the rods and pin not being in the plane of the counterweight is calculated by the method given in the second progress report, this becomes approximately an underbalance of 35 lb.

TABLE 6

Weights of Rotating and Reciprocating Parts, in Pounds—Light Santa Fe Type Locomotive of the A. T. & S. F. Ry.

RECIPROCATING PARTS

Piston	985
Crosshead	480
Union link and lower end of combination lever	65
48% main rod, weight on crosshead pin	470
	2000

$$\frac{.666 \times 2000}{5} = 267$$

ROTATING PARTS

Driver Number	5	4	Main	2	1
Two-thirds of weight of reciprocating parts	267	267	267	267	267
Main rod, weight on crank pin	.....	.....	510	.....	.....
Side rod, weight on crank pin	125	302	762	314	125
Crank pin	70	70	390	70	67
Crank pin hub	157	157	325	157	157
Total	619	796	2254	808	616
Equivalent weight required at center of counterweight	604	836	2365	848	602
Weight used	588	760	1495	802	579
Difference between weight required and weight obtained	—16	—76	—870	—46	—23
Equivalent difference at crank pin circle	—17	—72	—828	—44	—23
Overbalance or underbalance at crank pin circle for rotating parts only	+250	+195	—561*	+223	+244

*No information is at hand on the weight or position of the bobs or position of planes of the outside rotating parts.

TABLE 7

Weights of Rotating and Reciprocating Parts, in Pounds—Heavy Santa Fe Type Locomotives of the A. T. & S. F. Ry.

RECIPROCATING PARTS

Piston	1070
Crosshead	705
Union link and lower end of combination lever	62
40% main rod, weight on crosshead pin	585
	2422

$$\frac{.50 \times 2422}{4} = 303$$

It was not possible to counterbalance for the whole of the rotating weight in the main wheel; the deficiency of 21 lb. was distributed equally among the other drivers, bringing their total up to 308 lb.

ROTATING PARTS

Driver Number	5	4	Main	2	1
50% of weight of reciprocating parts	308	308	.....	308	308
Main rod, weight on crank pin	.....	.....	877	.....	.....
Side rod, weight on crank pin	172	493	785	454	166
Crank pin	118	130	500	130	105
Crank pin hub	144	144	271	144	144
One-half eccentric crank	.....	.....	91	.....	.....
Total	742	1075	2524	1036	723
Equivalent weight required at center of counterweight	574	899	2714	866	564
Weight used	574	899	2693	866	564
Difference between weight required and weight obtained	.....	.....	21	.....	.....
Equivalent difference at crank pin circle	.....	.....	20	.....	.....
Overbalance or underbalance at crank pin circle for rotating parts only	+308	+308	—20*	+308	+308

*If the effect of the rods and pin not being in the plane of the counterweight is calculated by the method given in the second progress report, this becomes approximately an underbalance of 360 lb.

TABLE 8

Weights of Rotating and Reciprocating Parts, in Pounds—Pacific Type Locomotive of the D. L. & W. R. R.

RECIPROCATING PARTS

Piston	961
Crosshead	585
Weight of main rod on crosshead pin	335
	1881

$$\frac{2 \times 1881}{3 \times 3} = 418$$

ROTATING PARTS

Driver Number	3	Main	1
Two-thirds of weight of reciprocating parts	418	418	418
Main rod, weight on crank pin	.....	627	.....
Side rod, weight on crank pin	159	517	145
Crank pin	116	398	114
Crank pin hub	113	226	113
One-half eccentric crank	.....	60	.....
Total	806	2246	790
Equivalent weight required at center of counterweight	478	1562	468
Weight used	478	1562	468
Difference between weight required and weight obtained	0	0	0
Equivalent difference at crank pin circle	0	0	0
Overbalance or underbalance at crank pin circle for rotating parts only	+418	+418*	+418

*If the effect of the rods and pin not being in the plane of the counterweight is calculated by the method given in the second progress report, this becomes approximately an overbalance of 175 lb.

TABLE 9

Weights of Rotating and Reciprocating Parts, in Pounds—Mikado Type Locomotive of the D. L. & W. R. R.

RECIPROCATING PARTS

Piston	1097
Crosshead	570
Weight of main rod on crosshead pin	405
	2072

$$\frac{2 \times 2072}{3 \times 4} = 345$$

ROTATING PARTS

Driver Number	4	Main	2	1
Two-thirds of weight of reciprocating parts	345	345	345	345
Main rod, weight on crank pin	.....	670	.....	.....
Side rod, weight on crank pin	130	392	308	135
Crank pin	83	406	77	67
Crank pin hub	135	210	135	127
One-half eccentric crank	.....	66	.....	.....
Total	693	2089	865	674
Equivalent weight required at center of counterweight	530	2089	661	515
Weight used	530	2089	661	515
Difference between weight required and weight obtained	0	0	0	0
Equivalent difference at crank pin circle	0	0	0	0
Overbalance or underbalance at crank pin circle for rotating parts only	+345	+345*	+345	+345

*If the effect of the rods and pin not being in the plane of the counterweight is calculated by the method given in the second progress report, this becomes approximately an overbalance of 95 lb.

In the tests made in California, the locomotive of the A. T. & S. F. Ry. used was the same as the Heavy Santa Fe type shown in Fig. 1 except that on account of the use of oil burners the weight on the trailer was only 22,900 lb. That of the S. P. Co. had nearly the same driver loads, but the load on the trailer was 29,400 lb. The Mikado type locomotive used on the preliminary tests on the I. C. R. R. was of the same class as used on that railroad in 1918. The Ten Wheel type was a much lighter locomotive.

5. **Conduct of the Test and the Reduction of the Data.**—The methods employed in the tests of track were the same as those used in the tests described in the first and second progress reports. Eight stremmatographs were used simultaneously. (A description of these instruments will be found in the first progress report.) Four stremmatographs were placed on one rail between ties at distances apart approximately equal to the average spacing of the drivers of the locomotive used on the test, and the other four instruments directly opposite on the other rail. The driving mechanism used rotated the discs of all the instruments simultaneously. The correlation of a point on the record of one of the discs with the point of any other disc at the same moment was possible. As the locomotive passed the test section a record of the strains in the rail was made on each instrument. The passage of each wheel of one side of the locomotive and tender thus was recorded on four instruments and each wheel of the other side on four other instruments. A run then gave records of what happened under one pair of wheels on eight instruments, and as each instrument holds two discs sixteen records in all were made. As usually operated, a disc would hold the records of four runs. Fresh discs would then be inserted.

The use of eight stremmatographs—four had been the largest number used in previous tests—had advantages even on the tests of straight track. The strain under companion drivers was recorded simultaneously; in fact, with just the right spacing the strain under four pairs of wheels was recorded at the same instant, thus giving the instantaneous effect of the four pairs of drivers. The distance from the first instrument to the fourth was generally about equal to the circumference of a driver; this facilitated finding the effect of counterbalance. The use of eight instruments was more likely to give the general effect of the locomotive as a whole at a given time than did the smaller number. The time taken to obtain an adequate amount of data was shortened, thus reducing the length of time the locomotive was taken from service.

The chief purpose in using eight instruments, however, was to facilitate the tests on curved track, where, of course, the stresses in the two rails are generally unequal. The use of two discs in each stremmatograph, one recording the strain at one edge of the base of rail and the other that at the other edge, permitted the measurement of the lateral bending strains in the rails of curved track. It is believed that these tests were the first to make a measurement of the bending stresses developed on

curved track. The means employed were entirely successful. The preliminary tests made to learn the precautions and methods necessary for the satisfactory conduct in the testing of curved track were made on a curve of a branch line. It will be noted that by the method used the stresses in the two edges of both inner and outer rail of the curved track, caused by both the vertical bending and the lateral bending of the rail, were measured under four pairs of drivers at about the same time.

Unless otherwise noted, in all runs steam was shut off as the locomotive approached the test section of track and the locomotive "coasted" past the instruments. The speeds were read from a speedometer in the cab connected with the tread of the trailer or the front wheel of the tender, the instrument having been checked up over a measured length of track. The locomotive was then backed over the track and the next run made. For each set of tests the order of speeds for consecutive runs was the same, varying from the lowest speed to the highest.

The position of the counterweight with respect to one instrument was observed for each run. It was found unnecessary to control the position of counterweight on different runs; the starting and stopping and the running over curves within the length of the runs gave sufficient distribution of the position of counterweight with respect to the instruments throughout the revolution of the driver.

Fig. 5 shows the position of the stremmatographs in one of the tests of curved track. Variations from this arrangement were made, but the diagram is representative of all of the tests. As all track was laid with alternate joints it was necessary that a rail joint lie within the test section. Generally the joint was between the two middle instruments, some-

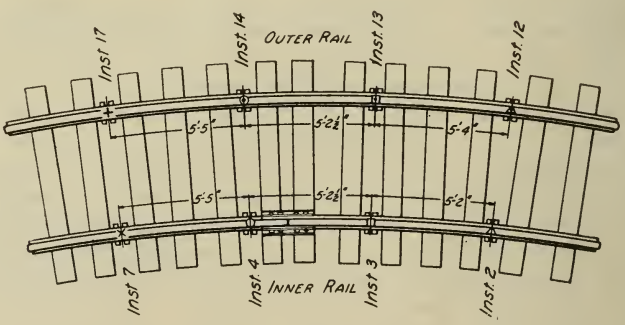


Fig. 5.—Position of the Eight Stremmatographs in a Test of Curved Track.

times on one rail and sometimes on the other. A study of the records of the several instruments failed to show any general characteristics of the strains recorded in the instrument nearest the rail joint.

The process followed in the reduction of the data obtained by the stremmatographs was the same as that which was described in the earlier reports. The stremmatograph records were read with a binocular microscope fitted with a micrometer eyepiece. Readings were taken for points in the record corresponding to a wheel over the instrument and to a point between wheels, the high point of the record being assumed to have been made when the wheel was directly over the instrument. In reducing these measurements the readings were multiplied by the proper microscopic constant and then reduced to stresses by multiplying by a constant which involves the position of the neutral axis of the rail section, the vertical distance of the needle bar below the base of rail, the modulus of elasticity of steel (taken as 30,000,000 lb. per sq. in.), and the gage length (which was four inches). A correction was also made to allow for the variation of moment and stress over the gage length to obtain the maximum stress at the middle of the gage length by multiplying the average stress over the gage length for wheel over instrument by the factor 1.04, as was done with the tests recorded in the first progress report, so that the stresses reported at points of positive moment are stresses in pounds per square inch at the base of the rail at the middle of the gage length. The variation over the gage length at points of negative moment was slight and no correction was used for the stresses at such points.

In general, on straight track the readings of the two discs of an instrument, giving stresses at the two edges of the base of the rail, were averaged and the average was taken as the observation for that instrument for the given run for the given position of wheel. Usually on straight track, in case the record of one disc of an instrument was defective and that on the companion disc was clear, the one good record was generally also discarded. In the case of curved track, all readable records were used, even though the record on the companion disc was defective. It was desired to keep the strains at the two edges separated. It was also found that on curved track there was less vibration in the rail and less opportunity for errors being introduced through the use of a record at only one edge. Besides, it was found that defective records were less frequent on curved track, usually less than 10 per cent of the points having to be discarded.

The accuracy of the records and their reduction is considered to be as good as that of the stremmatograph data reported in the two earlier reports.

The time required for the reduction of the data of the large number of observations made in all the tests was considerable. The reading of the stremmatograph records, the calculation of the data, and the representation of the results for study and interpretation involved a large amount of work, as may be judged from a consideration of the number of microscopic readings of strains made, approximately 470,000. With clear records skilled observers learned to make 1,000 to 1,200 such readings in a day.

Table 1 gives the properties of the sections of the rails new. Profiles were taken of the sections of the rail in the track. With the exception of the 105-lb. rail on the D. L. & W. R. R. the rails were not worn enough to have a sufficiently marked effect on the moment of inertia and section modulus about the horizontal axis to make it seem necessary to use the section modulus of the worn rail in the analytical calculation of stresses in rail. On the 7½° curve the moment of inertia and section modulus of the inner rail were reduced about 3 per cent and the outer rail about 6 per cent, but here also the properties of the new rail have been used in making comparisons. The effect of wear on the section modulus about a vertical axis would be but little, since the base of rail, which is not changed, forms the large factor in making up the section modulus. For lateral bending of the rail on curved track the properties of new rails were also used.

II. TESTS ON STRAIGHT TRACK

6. **Results of Tests.**—The tests on straight track, although made for the purpose of supplying a basis of comparison with results on curved track, furnish considerable additional information on the action of straight track, particularly on the effect of speed and counterbalance with several types of locomotives.

To determine the stiffness of track and the value of the modulus of elasticity of rail support *u*, measurements of track depression under the truck of loaded cars were made. For the significance of the term modulus of elasticity of rail support see the first progress report of the Committee on Stresses in Railroad Track.* The method of making the test was similar to that described in the first progress report, the level bar being used.

On the A. T. & S. F. Ry. the lighter of the two loads applied was an empty car, and the heavier load was a heavily loaded coal car. On the D. L. & W. R. R., the lightest load was an empty car, the medium load a loaded coal car, and the heaviest load a coal car heavily loaded with ore. In each case the load considered was that on one truck at an end of the car. The load on the wheels of the truck used was carefully weighed, a check on the weight being obtained by also weighing the load on the other truck and the load on the whole car.

Fig. 6 and 7 give the results of test of track depression on the track of A. T. & S. F. Ry. at Ribera, New Mexico, and Fig. 8 and 9 are for tests on the track of the D. L. & W. R. R. at Dover, New Jersey. Location A on the A. T. & S. F. Ry. covered the location of the stremmatographs on the regular tests at Ribera; Location B was 130 feet east of Location A. Location C coincided with the location of the stremmatographs at Dover; Location D was 260 feet east of Location C. The track depression profiles indicate that the track on which the tests were made was in fairly good condition, the "play curve" showing uniformity of play

*Trans. Am. Soc. C. E., 82, p. 1204; Proceedings A.R.E.A., Vol. 19, p. 884.

between rail and tie and tie and ballast, and uniformity of stiffness in each case.

The value of the modulus of elasticity of rail support u, calculated from these track depression profiles by the method described in the first progress report, was 1,600 lb. per lin. in. of rail, for the track on both the A. T. & S. F. Ry. and the D. L. & W. R. R. These values indicate a fairly stiff track.

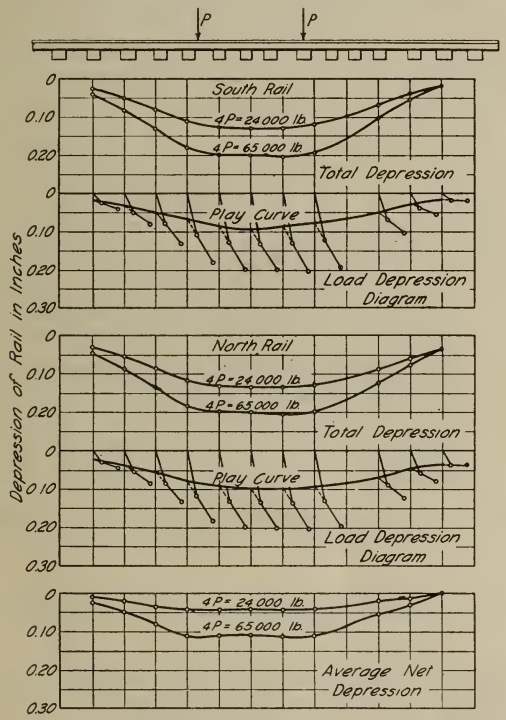


Fig. 6.—Track Depression Profiles, Static Load Tests on the A. T. & S. F. Ry., Location A.

In all the tests to find the stresses in the rail four stremmatographs were placed on one rail and four on the other, the four instruments on one rail being spaced at distances of about 66 inches, and the set on the other rail being placed directly opposite. For all stress diagrams on straight track the stress given is the mean stress in base of rail; that is, the average of the stresses at the outside and inside edges of the rail, and the stress there recorded is the average of the results on both rails as found from the records of the eight instruments. In Figs. 10, 11 and 12 are given the mean stress in the base of rail for the high and the low

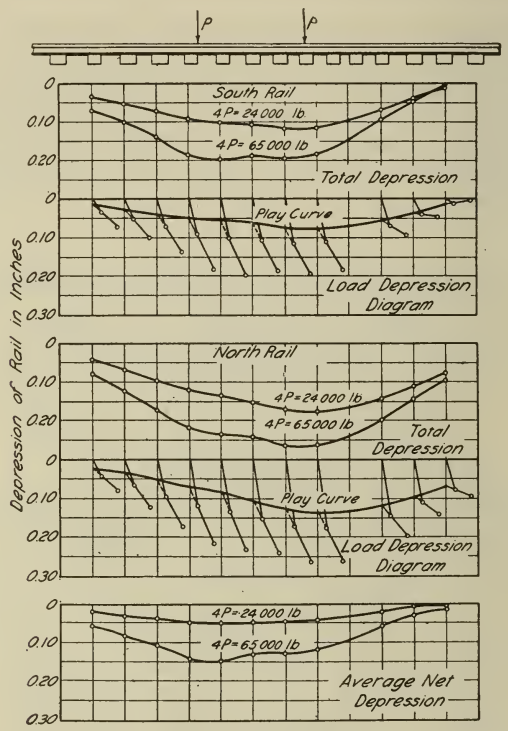


FIG. 7—TRACK DEPRESSION PROFILES, STATIC LOAD TESTS ON THE A. T. & S. F. RY., LOCATION B.

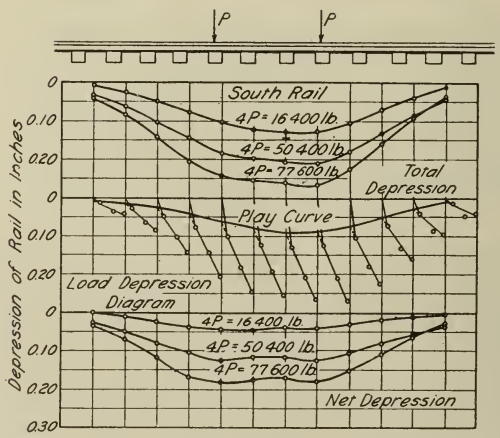


FIG. 8.—TRACK DEPRESSION PROFILES, STATIC LOAD TESTS ON THE D. L. & W. R. R., LOCATION C.

position of the counterweight for the Pacific type locomotive, the Light Prairie type, and the Balanced Compound Prairie type, respectively, in the tests of track of the A. T. & S. F. Ry. at East Fort Madison, Illinois. In Fig. 13, 14, 15, 16, 17, and 20 are given the corresponding stresses for the tests with the Mountain type locomotive coasting down-grade, the Mountain type running up-grade, the Light Santa Fe type, and the Heavy Santa Fe type coasting down-grade, the Heavy Santa Fe type working steam down-grade and the Double Trailer Heavy Santa Fe type coasting down-grade, respectively, at Ribera, New Mexico.

In Tables 10 to 18 are given the average or mean values of the stresses in base of rail under the locomotive wheels found in the tests, and also the values at the high point and the low point of the counterweight.

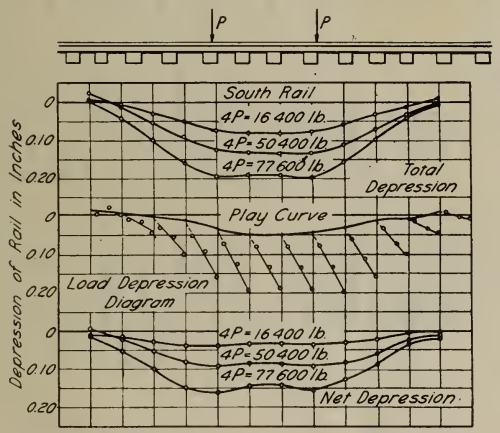


Fig. 9.—Track Depression Profiles, Static Load Tests on the D. L. & W. R. R., Location D.

The calculated stresses under the nominal static wheel loads are also given. The calculations are based on the method of analysis given in the first progress report, and the value of u found from the tests of track depression was used. It will be noted that the calculated values of the stresses for static loads in general do not differ greatly from the observed values at a speed of 5 miles per hour. Where there is a noticeable difference, it seems probable that the actual distribution of load among the individual wheels is not the same as the assumed distribution. It is found, however, that the average of the calculated stress under the several wheels of the locomotive is almost exactly the same as the average of the observed stresses under the wheel at a speed of 5 miles per hour, the greatest difference in all the tests being about 6 per cent.

Attention should be called to the fact that the stresses given in the diagrams and tables are average stresses, the result of averaging a num-

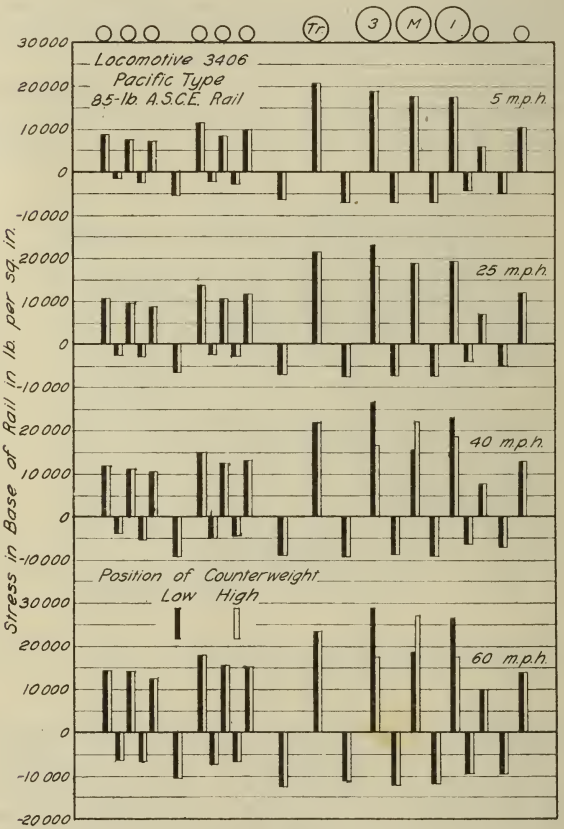


Fig. 10.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5450-58, Pacific Type Locomotive of the A. T. & S. F. Ry.

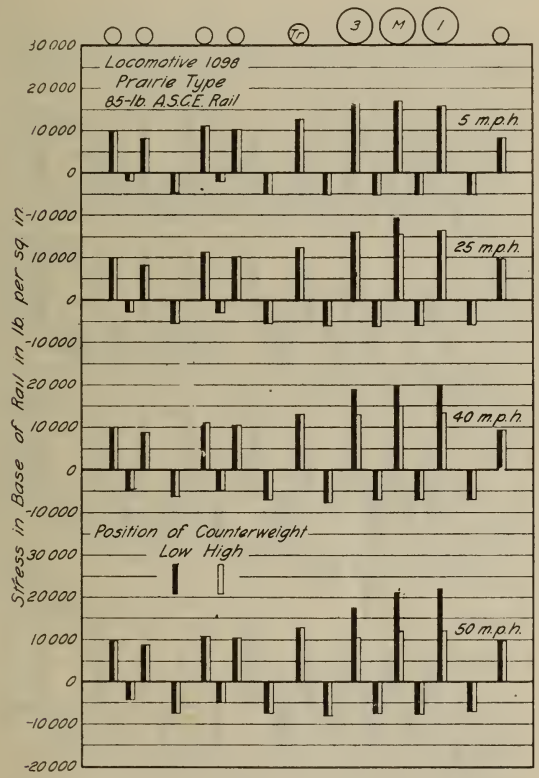


Fig. 11.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5441-49, Light Prairie Type Locomotive of the A. T. & S. F. Ry.

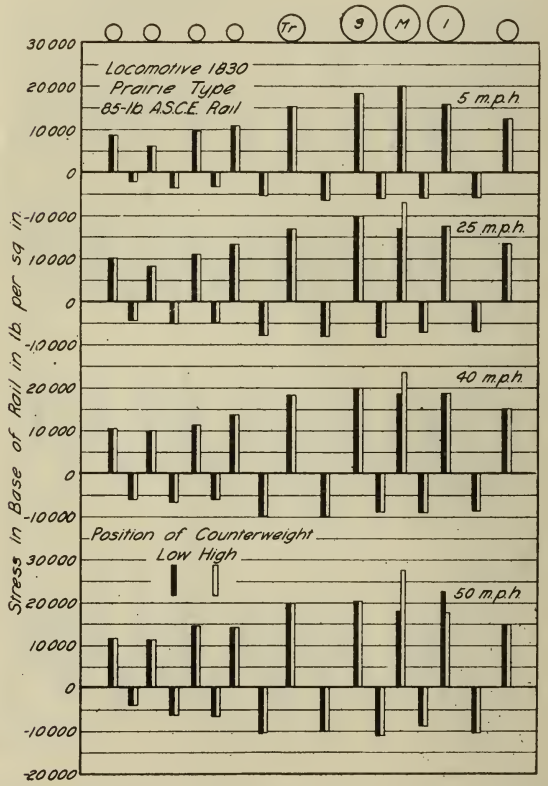


Fig. 12.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5434-40, Balanced Compound Prairie Type Locomotive of the A. T. & S. F. Ry.

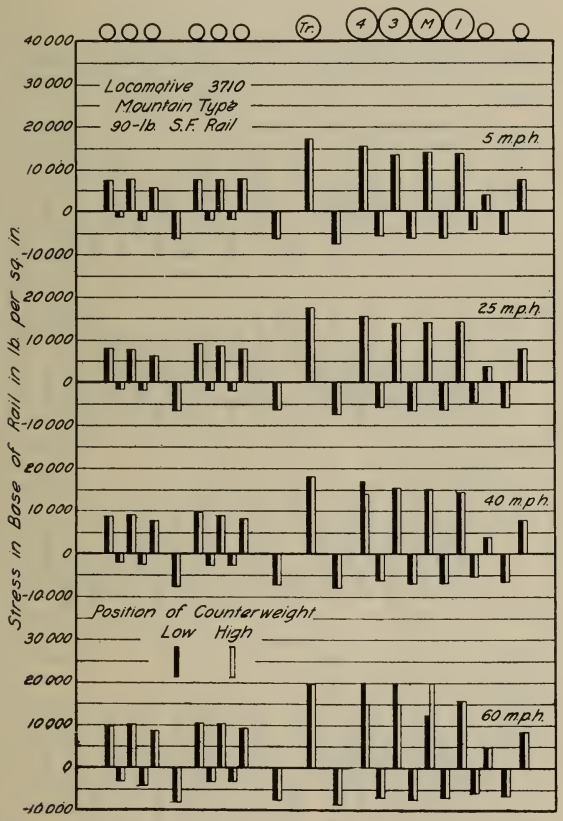


Fig. 13.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5338-44, Mountain Type Locomotive of the A. T. & S. F. Ry.

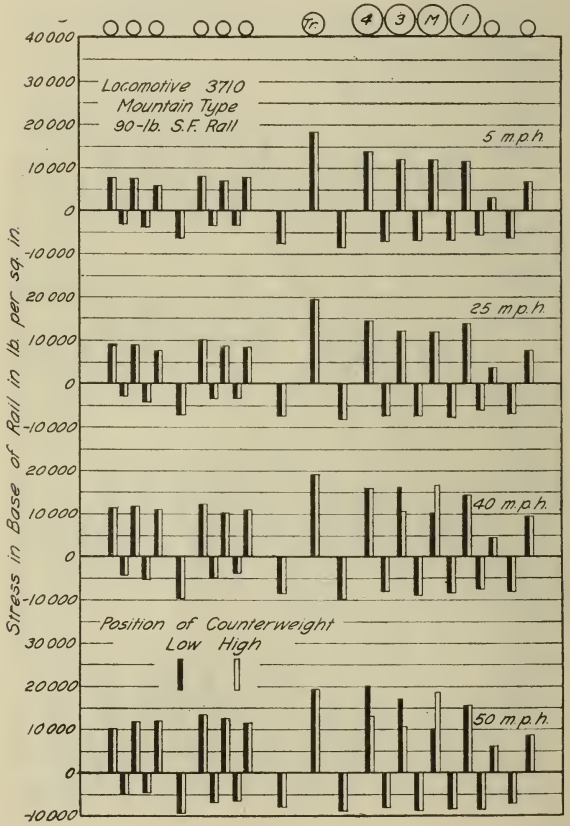


Fig. 14.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5345-50, Mountain Type Locomotive of the A. T. & S. F. Ry.

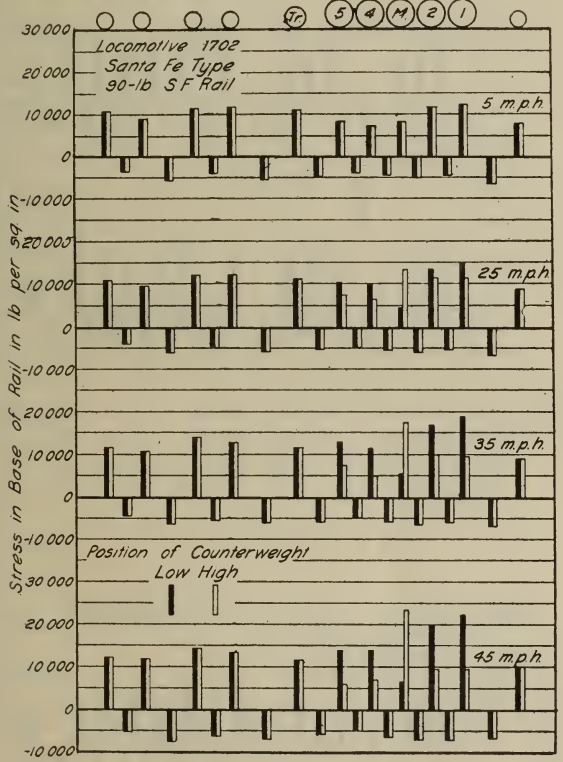


Fig. 15.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5300-18, Light Santa Fe Type Locomotive of the A. T. & S. F. Ry.

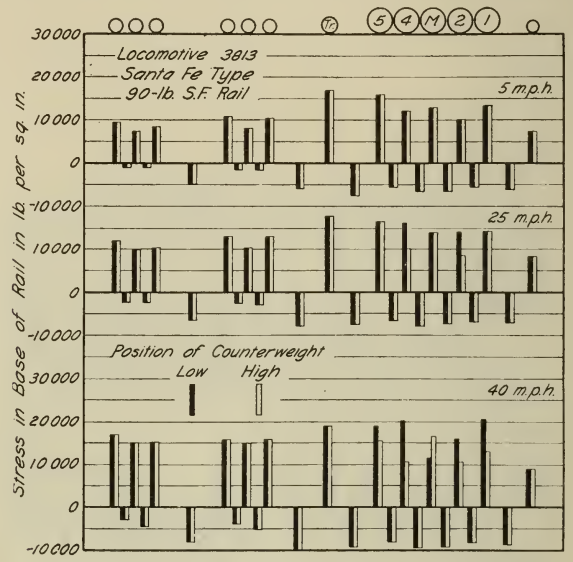


Fig. 16.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5401-08, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

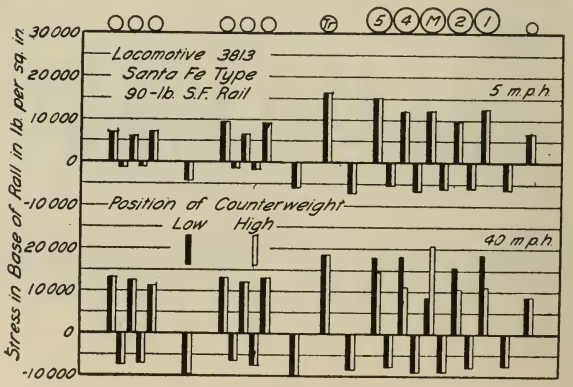


Fig. 17.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5409-15, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

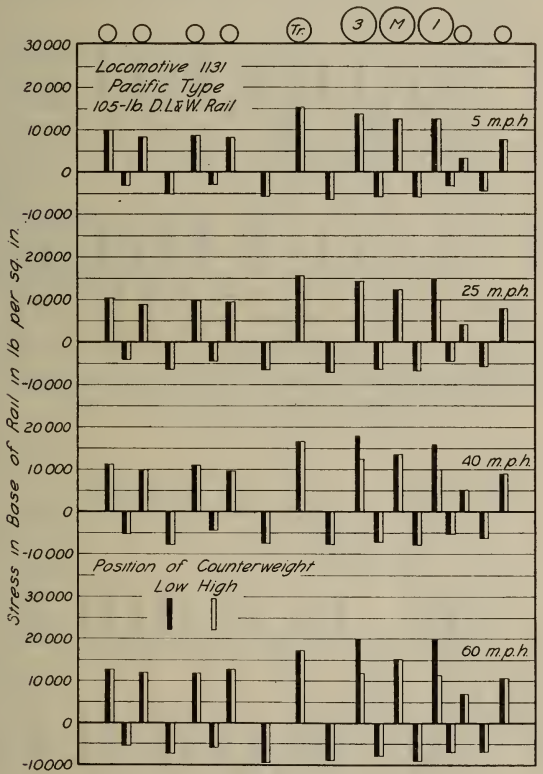


Fig. 18.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5136-49, Pacific Type Locomotive of the D. L. & W. R. R.

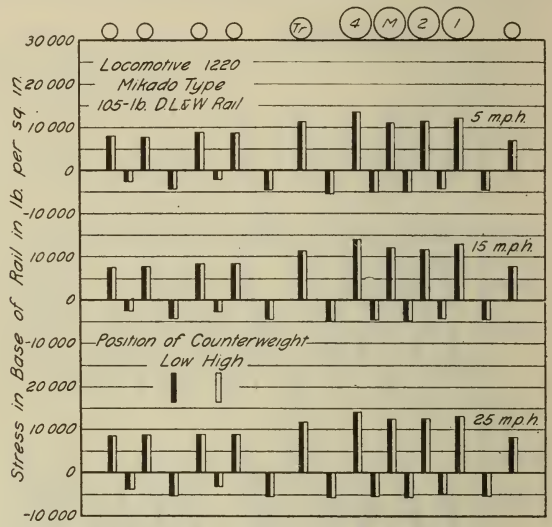


Fig. 19.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5206-20, Mikado Type Locomotive of the D. L. & W. R. R.

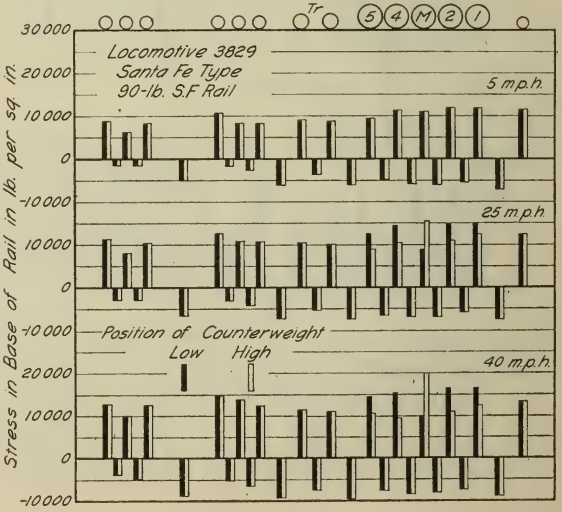


Fig. 20.—Stress in Rail on Straight Track at High and Low Position of Counterweight—Series 5416-24, Double Trailer Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

TABLE 10

Stresses in Rail with Pacific Type Locomotive of the A. T. & S. F. Ry.—Maximum, Minimum, and Mean Values from Curves for Series 5450-58.

Stresses are given in pounds per square inch at base of rail.

Speed in m. p. h.	Position of Counterweight	Trailer	Driver Number			Front Truck Wheels	
			3	Main	1	2	1
5	Mean value	20,700	18,900	17,700	17,500	6,000	10,400
25	Up		18,000				
	Down		23,000				
	Mean value	21,400	20,200	18,800	19,200	6,900	11,800
40	Up		16,500	22,000	18,500		
	Down		26,500	15,500	23,000		
	Mean value	21,800	21,000	19,100	20,500	7,700	12,800
60	Up		17,500	27,000	17,500		
	Down		29,000	18,500	26,500		
	Mean value	23,400	23,500	23,100	22,200	9,800	13,800
Calculated Stress under static load		22,900	19,600	15,000	17,000	4,600	9,400
Calculated additional stress due to counterbalance at 60 m. p. h.			+5,900	+4,700	+5,900		

TABLE 11

Stresses in Rail with Light Prairie Type Locomotive of the A. T. & S. F. Ry.—Maximum, Minimum and Mean Values from Curves for Series 5441-49.

Stresses are given in pounds per square inch at base of rail.

Speed in m. p. h.	Position of Counterweight	Trailer	Driver Number			Front Truck Wheel
			3	Main	1	
5	Mean value	12,600	16,400	16,900	15,800	8,200
25	Up			15,500		
	Down			19,500		
	Mean value	12,400	16,100	17,300	16,400	9,600
40	Up		13,000	15,000	13,500	
	Down		19,000	20,000	20,000	
	Mean value	13,100	16,100	16,700	15,700	9,300
50	Up		10,500	12,000	12,000	
	Down		17,500	21,000	22,000	
	Mean value	12,700	14,200	16,400	16,800	9,700
Calculated Stress under static load		14,200	13,700	14,000	14,300	10,000

ber of observations, usually a large number, from 50 to 125. This means that many observations lie above the average value and many below. In fact, when all the observations are plotted for a revolution of a driver there is a belt of points within two rather definite boundaries, one on each side of the line of averages, as is illustrated in Fig. 21 to 25. Within these boundaries the points are fairly well distributed, and outside of them a few scattered points will be found. For straight track the boundaries of the belt of points may be 5,000 lb. per sq. in. above and below the average line.

From an inspection of the diagrams and tables it will be noted that at a speed of 5 miles per hour the average stress in the base of rail of the 85-lb. A.S.C.E. section ranged from 15,800 to 20,000 lb. per sq. in. under the drivers of the Pacific type locomotive and the two Prairie types of the

TABLE 12

Stresses in Rail with Balanced Compound Prairie Type Locomotive of the A. T. & S. F. Ry.—Maximum, Minimum and Mean Values from Curves for Series 5434-40.

Stresses are given in pounds per square inch at base of rail.

Speed in m. p. h.	Position of Counterweight	Trailer	Driver Number			Front Truck Wheel
			3	Main	1	
5	Mean value	15,300	18,400	20,000	15,800	12,500
25	Up			23,000*		
	Down			17,000*		
	Mean value	16,800	19,800	20,200	17,400	13,500
40	Up			23,500*		
	Down			18,500*		
	Mean value	18,200	19,800	20,000	18,700	15,100
50	Up			27,500*	17,500	
	Down			18,000*	22,500	
	Mean value	19,700	20,200	21,300	20,000	14,800
Calculated Stress under static load		14,700	18,800	15,100	18,700	11,200

*The maximum and minimum occur at the quarter points.

TABLE 13

Stresses in Rail with Mountain Type Locomotive of the A. T. & S. F. Ry.—Maximum, Minimum, and Mean Values from Curves for Series 5338-44.

Stresses are given in pounds per square inch at base of rail.

Speed in m.p.h.	Position of Counterweight	Trailer	Driver Number				Front Truck Wheels	
			4	3	Main	1	2	1
5	Mean value	17,000	15,300	13,300	14,000	13,600	4,000	7,500
25	Up							
	Down							
	Mean value	17,400	15,400	13,900	14,100	14,200	3,800	8,000
40	Up		14,000					
	Down		17,000					
	Mean value	18,200	15,600	15,400	15,200	14,400	4,000	7,900
60	Up		15,000	15,000	20,000			
	Down		20,000	20,000	12,500	*		
	Mean value	19,500	17,200	17,300	17,100	15,900	5,000	8,500
Calculated Stress under static load		21,200	14,400	12,100	11,400	14,000	2,800	7,300
Calculated additional stress due to counterbalance at 60 m.p.h.			+4,700	+3,500	+3,500	+4,700		

*Not sufficient tests to determine the stress due to counterbalance.

A. T. & S. F. Ry. On the 90-lb. S. F. rail the average stress ranged from 10,100 to 15,900 lb. per sq. in. under the drivers of the Mountain type locomotive and Heavy Santa Fe types at 5 miles per hour. For the

Light Santa Fe type, having much lighter wheel loads, the corresponding stresses in the 90-lb. S. F. rail at the same speed range from 7,100 to 12,100 lb. per sq. in. For the Pacific and Mikado types of the D. L. & W. R. R., the stresses in the 105-lb. D. L. & W. section under the drivers

TABLE 14

Stresses in Rail with Light Santa Fe Type Locomotive of the A. T. & S. F. Ry.—Maximum, Minimum and Mean Values from Curves for Series 5300-18.
Stresses are given in pounds per square inch at base of rail.

Speed in m.p.h.	Position of Counterweight	Trailer	Driver Number					Front Truck Wheel
			5	4	Main	2	1	
5	Mean value	10,700	8,100	7,100	8,100	11,500	12,100	7,800
25	Up	.....	7,500	6,500	13,500	11,500	11,500	.....
	Down	.....	10,500	10,000	4,500	13,500	15,000	.....
	Mean value	11,300	9,500	7,900	9,900	12,600	13,100	8,800
35	Up	.....	7,500	5,000	17,500	10,000	19,000	.....
	Down	.....	13,000	11,500	5,500	17,000	9,500	.....
	Mean value	11,600	10,500	8,500	11,500	13,600	14,600	9,200
45	Up	.....	6,000	7,000	23,500	9,500	9,500	.....
	Down	.....	14,000	14,000	6,500	20,000	22,500	.....
	Mean value	11,700	10,200	10,500	14,300	14,600	14,800	10,100
Calculated Stress under static load		4,900	10,800	7,900	10,000	8,100	12,200	7,600

TABLE 15

Stresses in Rail with Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.—Maximum, Minimum and Mean Values from Curves for Series 5401-08.

Stresses are given in pounds per square inch at base of rail.

Speed in m.p.h.	Position of Counterweight	Trailer	Driver Number					Front Truck Wheel
			5	4	Main	2	1	
5	Mean value	16,900	15.900	12,200	12,900	10,100	13,400	7.300
25	Up	.....	.....	10,000	.....	8,500	.....	.....
	Down	.....	.....	16,000	.....	14,000	.....	.....
	Mean value	17,700	16,400	13,600	13,800	11,100	14,200	8,300
40	Up	.....	15,500	10,500	16,500	10,500	13,000	.....
	Down	.....	19,000	20,000	11,500	16,000	20,500	.....
	Mean value	18,900	17,200	14,700	14,200	12,700	16,300	8,800
Calculated Stress under static load		19,700	15,400	11,300	11,000	10,600	14,400	6,700
Calculated additional stress due to counterbalance at 40 m.p.h.		.....	+3,700	+3,600	−2,800	+3,600	+3,700	.....

at 5 miles per hour ranged from 11,100 to 13,700 lb. per sq. in. It should be borne in mind that the stress named above in every case is the average value found in a considerable number of runs and that when a locomotive is run over the track many times individual stresses will be found under a given wheel that are considerably greater in magnitude than the average

stress named. It should also be remembered that the lateral bending of the rail under the action of the locomotive on straight track will result in stresses at the outside edge or the inside edge of the base of rail that

TABLE 16

Stresses in Rail with Double Trailer Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.—Maximum, Minimum and Mean Values from Curves for Series 5416-24.

Stresses are given in pounds per square inch at base of rail.

Speed in m.p.h.	Position of Counterweight	Trailer		Driver Number					Front Truck Wheel
		2	1	5	4	Main	2	1	
5	Mean value.....	9,200	8,900	9,500	11,400	11,100	11,900	11,900	11,700
25	Up.....	.....	.....	9,000	10,500	15,500	11,000	12,500	.....
	Down.....	.....	.....	12,500	14,500	9,000	15,000	15,000	.....
	Mean value.....	10,400	10,100	10,100	12,400	12,200	13,000	13,900	12,400
40	Up.....	.....	.....	10,500	9,500	20,000	11,000	12,500	.....
	Down.....	.....	.....	14,500	15,500	10,000	16,500	16,500	.....
	Mean value.....	11,500	11,100	12,400	12,800	12,900	14,000	14,900	13,300
Calculated Stress under static load.....		8,700	7,500	12,600	9,500	10,000	11,300	14,500	8,600
Calculated additional stress due to counterbalance at 40 m.p.h....		.....	.....	+3,700	+3,600	—2,800	+3,600	+3,700	.....

TABLE 17

Stresses in Rail with Pacific Type Locomotive of the D. L. & W. R. R.—Maximum, Minimum, and Mean Values from Curves for Series 5136-49.

Stresses are given in pounds per square inch at base of rail.

Speed in m.p.h.	Position of Counterweight	Trailer	Driver Number			Front Truck Wheels	
			3	Main	1	2	1
5	Mean value.....	15,300	13,700	12,600	12,600	3,300	7,700
25	Up.....	.....	.....	.....	10,000	.....	.....
	Down.....	.....	.....	.....	15,000	.....	.....
	Mean value.....	15,700	14,300	12,400	12,300	4,200	7,900
40	Up.....	.....	12,500	.....	10,000	.....	.....
	Down.....	.....	18,000	.....	16,000	.....	.....
	Mean value.....	16,600	15,000	13,600	13,000	5,200	9,000
60	Up.....	.....	12,000	.....	11,500	.....	.....
	Down.....	.....	20,000	.....	20,000	.....	.....
	Mean value.....	17,400	15,800	15,300	15,700	7,100	10,900
Calculated Stress under static load.....		15,800	14,800	12,200	14,600	1,500	5,400
Calculated additional stress due to counterbalance at 60 m.p.h.....		.....	+5,900	+4,500	+5,900	.....	.....

are materially higher than the mean of the stresses at the two edges, and also that the effect of speed and counterbalance is to add considerably to the values found at 5 miles per hour.

7. **Effect of the Locomotive Counterbalance.**—In the Second Progress Report of the Joint Committee on Stresses in Railroad Track,* the problem of counterbalancing the locomotive is briefly considered. For clearness of expression the term counterweight is there and here used

TABLE 18

Stresses in Rail with Mikado Type Locomotive of the D. L. & W. R. R.—Maximum, Minimum, and Mean Values from Curves for Series 5206-20.

Stresses are given in pounds per square inch at base of rail.

Speed in m.p.h.	Position of Counterweight	Trailer	Driver Number				Front Truck Wheel
			4	Main	2	1	
5	Mean value	11,400	13,600	11,100	11,500	12,200	7,000
15	Up						
	Down						
	Mean value	11,400	14,100	12,000	11,900	12,000	7,800
25	Up						
	Down						
	Mean value	11,900	14,200	12,500	12,700	13,100	8,200
Calculated Stress under static load		12,600	12,200	10,100	10,100	13,000	6,500

as the weight applied or added to the driver at a point opposite the crank pin for the purpose of balancing or helping to balance the rotating and reciprocating parts, and the term counterbalance as the general condition of underbalance or overbalance as it exists in a given driver, while counterbalancing is used to refer to the general problem of endeavoring to balance the rotating and reciprocating parts.

The vertical effect of a lack of balance of the rotating parts is to increase or decrease the pressure of the drivers on the rails and to vary the pressure upward on the equalizing levers and the frame of the locomotive. Since the counterweight can not be made to neutralize both the vertical effect and the horizontal effect of the moving parts (part of them having horizontal motion only) a compromise must be made in the counterbalancing. It is the common practice in locomotive design to counterbalance for all rotating parts and for a portion of the reciprocating parts, this practice resulting in overbalancing the rotating parts so far as the vertical pressure on the track is concerned.

In general it may be considered as sufficiently near the truth to count the vertical dynamic augment produced by the excess or deficiency of counterbalance as a vertical pressure on the rail at each driver, with whatever further allowance may be necessary to provide for the counterweight not being in the same plane as the rotating parts which are to be counterbalanced. This method is fairly accurate because the change in depression of the track due to the added or decreased pressure on the rail is small in comparison with the total amount of depression of the locomotive spring produced by the weight of the locomotive tributary to the

*Trans. Am. Soc. C. E., Vol. 83, p. 1415, Proceedings A.R.E.A., Vol. 21, p. 652.

given driver. In making calculations parts of the side rod and connecting rod are considered as rotating parts. It seems proper then in making calculations to consider the vertical dynamic augment of each driver as being separate downward or upward loads which may be added algebraically to the individual driver loads on the rail.

The formula generally used for calculating the dynamic augment is the formula for centrifugal force:

$$C = \frac{W\,v^2}{g\,r} \qquad (12)$$

where W is the weight of the rotating or reciprocating part for which the dynamic augment is desired or the unbalanced part of the rotating parts in case the resulting rotational effect is wanted.

g is the acceleration of gravity.

v is the rotational velocity or speed of the part.

r is the radius of the circle described by the center of gravity of the rotating part.

It is convenient to transfer all weights to the equivalent weight at the center of crank pin and to use the radius of the circle taken by the center of crank pin relatively to the center of the driver.

For determining the vertical dynamic augment W should include the excess or deficiency of the rotating parts, and should not include the reciprocating parts, a portion of the main rod being taken as rotating part.

Attention should be called to the need for taking into account the condition that the center of gravity of the rods and other rotating parts is not in the same vertical plane as the counterweight and the rail. As shown in the second progress report this condition has a considerable effect upon the counterbalancing of the main drivers, but it is not generally important with the other drivers. A main driver with a seeming overbalance for the rotating parts as judged by the method of calculation generally used in design may really be an underbalanced wheel, while for an underbalanced main driver the deficiency in counterweight may be considerably greater than that expected when this eccentricity of position is not taken into account.

Sample cases of observed values of the stresses in rail throughout the revolution of a driver are plotted in Fig. 21, 22, 23, 24 and 25. The ordinates of the diagrams represent stresses at the base of the rail in pounds per square inch. The abscissas represent the position of the counterweight at the time the wheel passes over the instrument with respect to its down position, the scale being in fractions of a complete revolution of the wheel. Results given by the several instruments are indicated by individual symbols. The curves representing the stress for a given speed throughout an entire revolution were formed by first averaging the values of the plotted points in groups for each tenth of a revolution and then drawing a curve to represent as well as possible the position and trend of these points. As may be expected, at a speed of 5 miles per hour the variation in stress due to other causes is so much

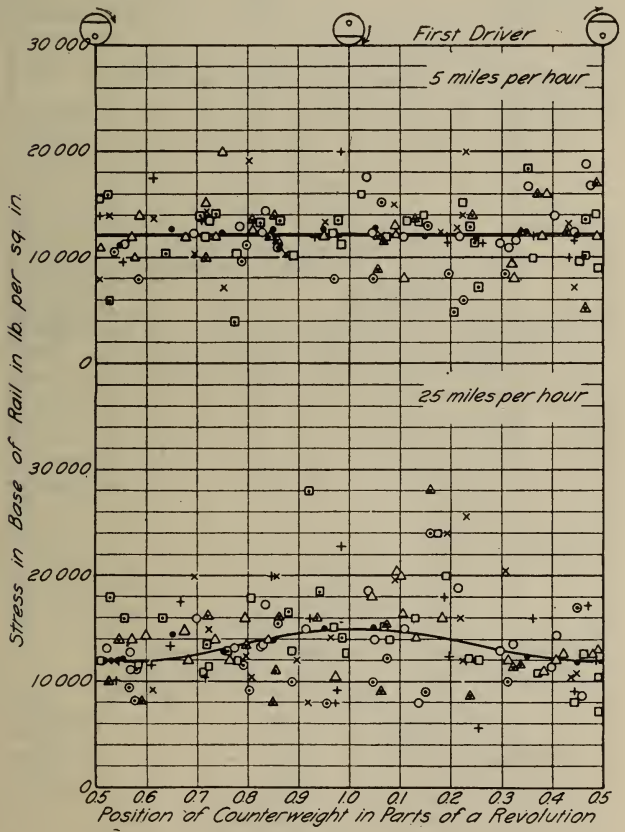


FIG. 21.—OBSERVED VALUES OF STRESS IN RAIL ON STRAIGHT TRACK—SERIES 5300-18, LIGHT SANTA FE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

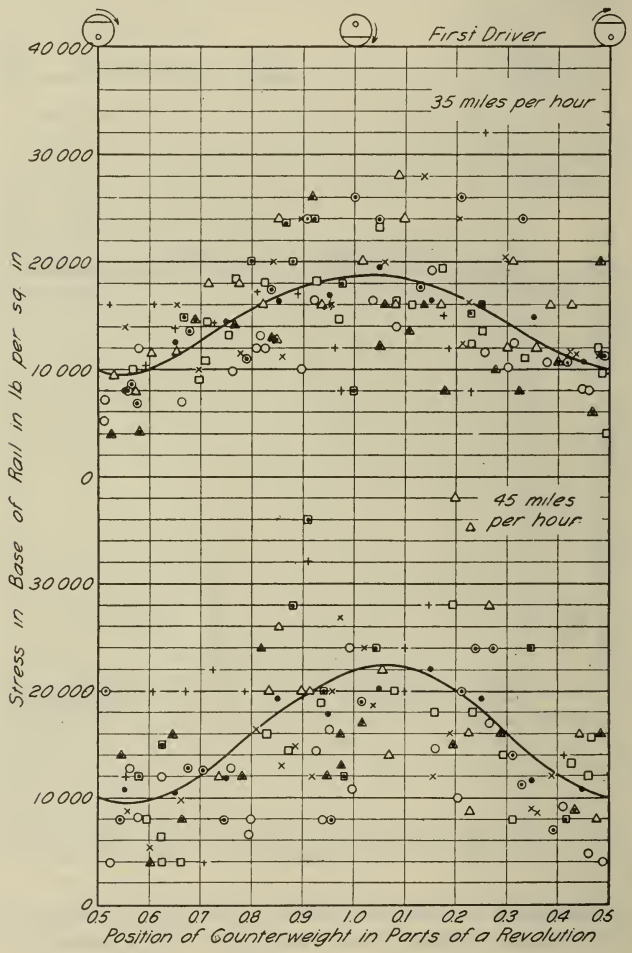


FIG. 22.—OBSERVED VALUES OF STRESS IN RAIL ON STRAIGHT TRACK—SERIES 5300-18, LIGHT SANTA FE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

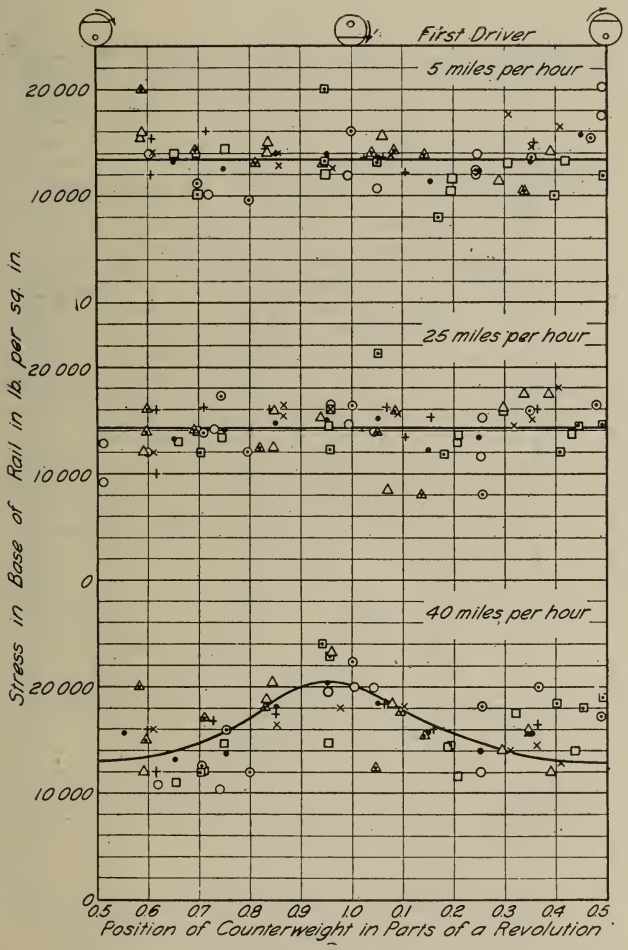


FIG. 23.—OBSERVED VALUES OF STRESS IN RAIL ON STRAIGHT TRACK—SERIES 5401-08, HEAVY SANTA FE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

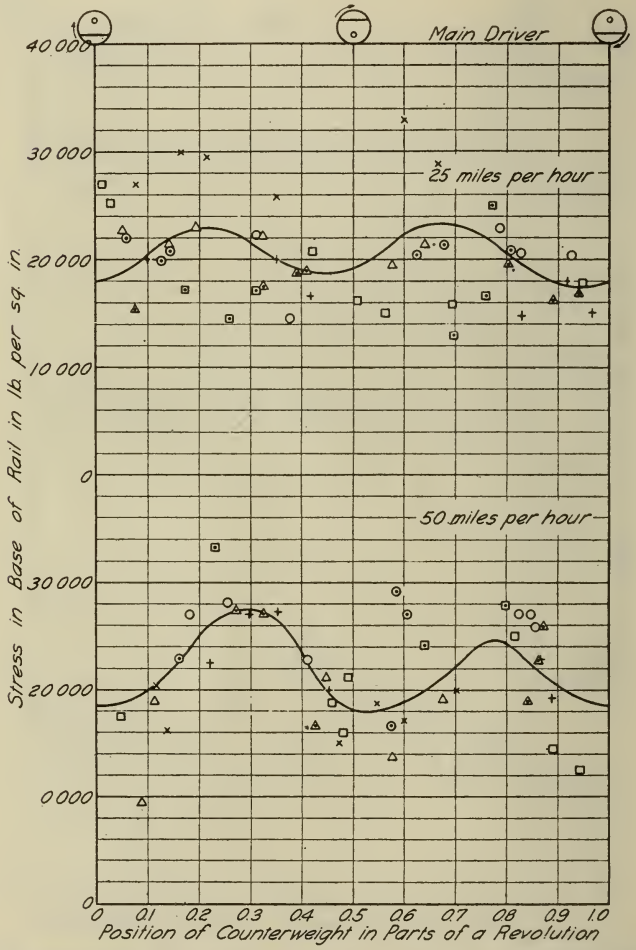


FIG.. 24.—OBSERVED VALUE OF STRESS IN RAIL ON STRAIGHT TRACK—SERIES 5434-40, BALANCED COMPOUND PRAIRIE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

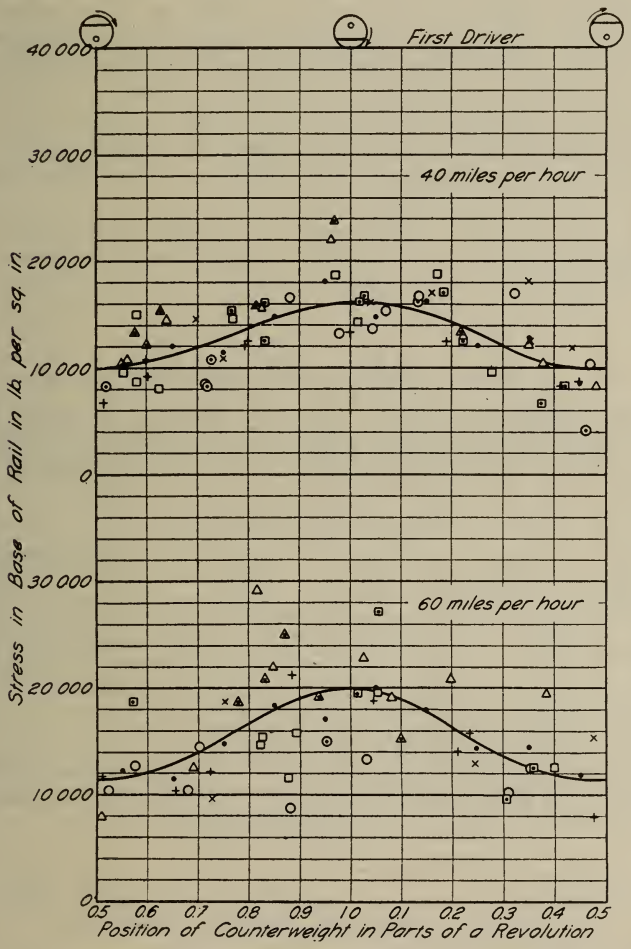


FIG. 25.—OBSERVED VALUES OF STRESS IN RAIL ON STRAIGHT TRACK—SERIES 5136-49, PACIFIC TYPE LOCOMOTIVE OF THE D. L. & W. R. R.

greater than that due to the effect of counterbalance as to mask any effect of counterbalance, and accordingly a straight line average of all the points has been used for all the wheels for a speed of 5 miles per hour. The average of all the observed values, which is approximately the average ordinate for the curves, is called the mean value at the given speed. The difference between this mean value and the high point of the average curve may be taken to be the effect of the counterbalance in increasing the stress (which may be due to overbalance or to underbalance), and the difference in the value to the low point of the curve represents the effect with the counterweight in the opposite position. The position of the counterweight for the maximum and minimum values is also represented on the diagrams.

The same method of determining the average curve of the stress throughout the revolution of a driver was used for all of the tests. Fig. 26 to 32 give the curves so found.

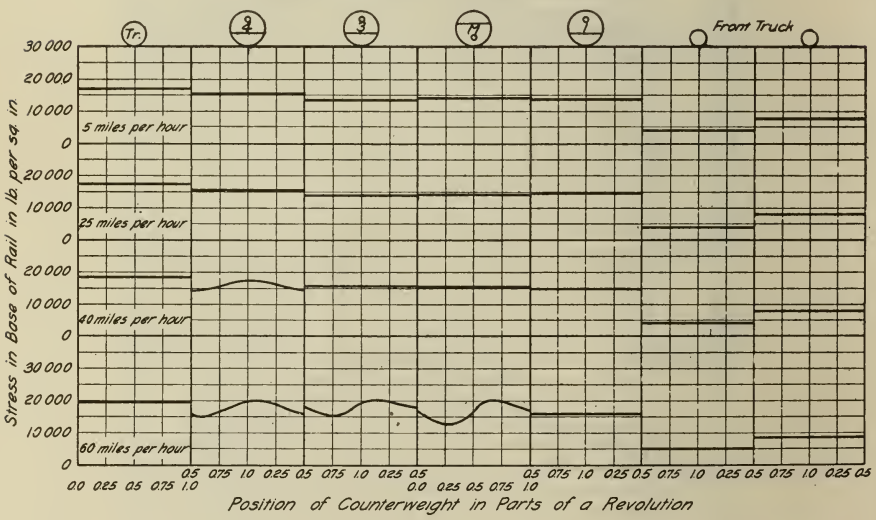


Fig. 26.—Curves of Average Stress in Rail on Straight Track Throughout Revolution of Driver—Series 5338-44, Mountain Type Locomotive of the A. T. & S. F. Ry.

Since the observations were well distributed throughout the revolution of a driver the average of the observed values under any wheel at any speed may be taken as the average stress in the rail at that speed without reference to counterbalance effect. The difference between this average stress and the high point and low point of the curve is taken to be the effect of counterbalance.

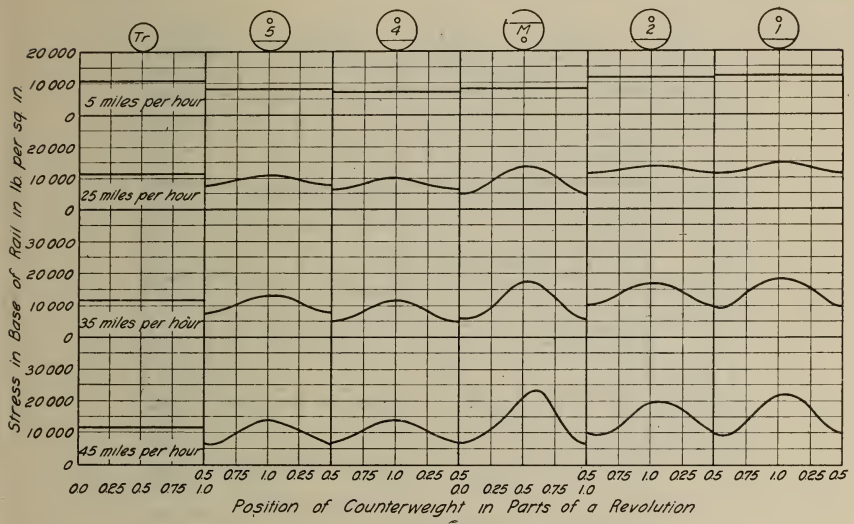


FIG. 27.—CURVES OF AVERAGE STRESS IN RAIL ON STRAIGHT TRACK THROUGHOUT REVOLUTION OF DRIVER—SERIES 5300-18, LIGHT SANTA FE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

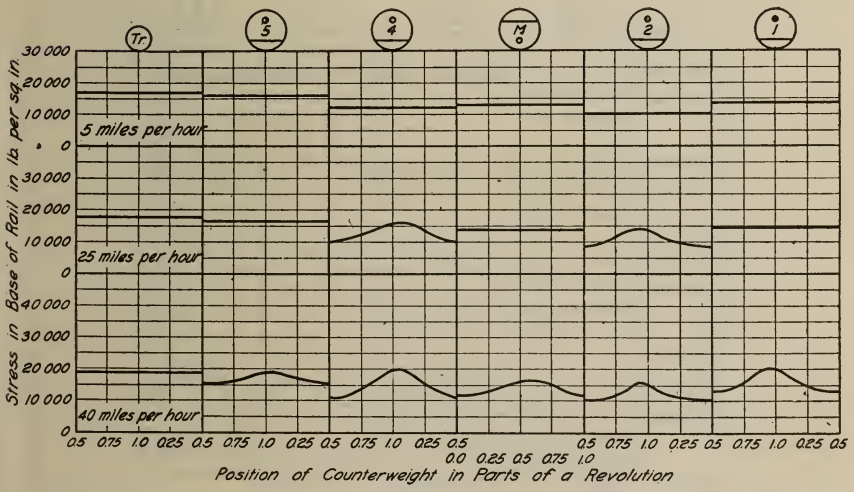


FIG. 28.—CURVES OF AVERAGE STRESS IN RAIL ON STRAIGHT TRACK THROUGHOUT REVOLUTION OF DRIVER—SERIES 5401-08, HEAVY SANTA FE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

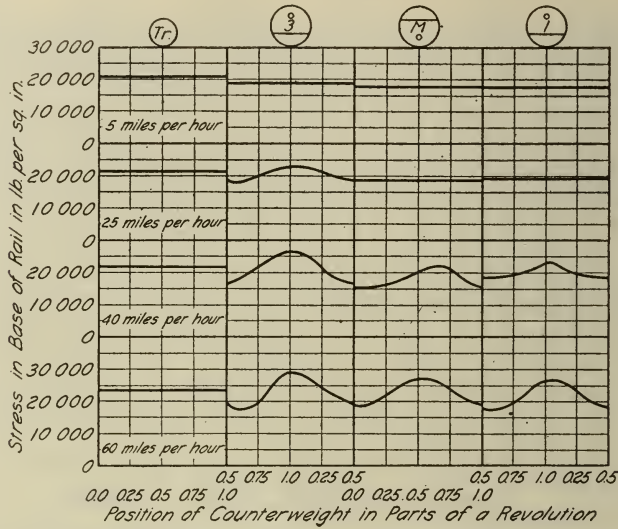


FIG. 29.—CURVES OF AVERAGE STRESS IN RAIL ON STRAIGHT TRACK THROUGHOUT REVOLUTION OF DRIVER—SERIES 5450-58, PACIFIC TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

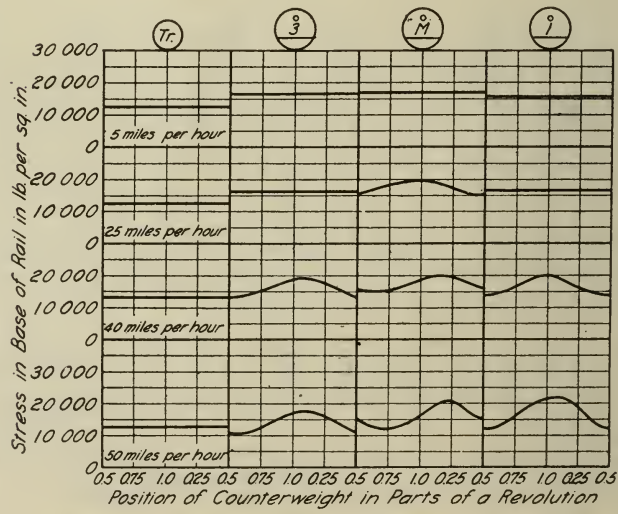


FIG. 30.—CURVES OF AVERAGE STRESS IN RAIL ON STRAIGHT TRACK THROUGHOUT REVOLUTION OF DRIVER—SERIES 5441-49, LIGHT PRAIRIE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

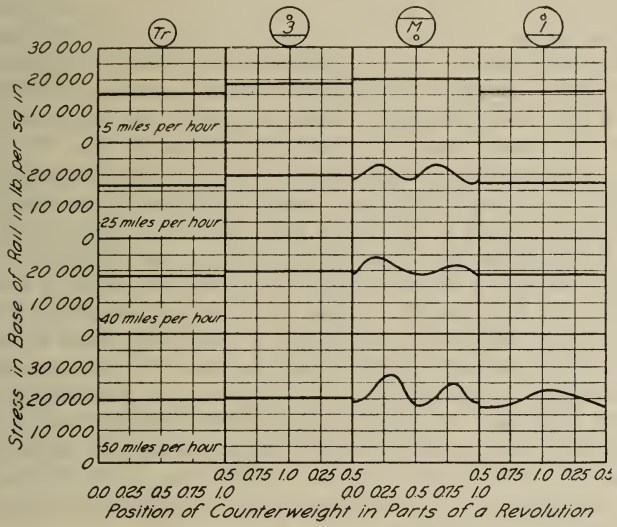


FIG. 31.—CURVES OF AVERAGE STRESS IN RAIL ON STRAIGHT TRACK THROUGHOUT REVOLUTION OF DRIVER—SERIES 5434-40, BALANCED COMPOUND PRAIRIE TYPE LOCOMOTIVE OF THE A. T. & S. F. RY.

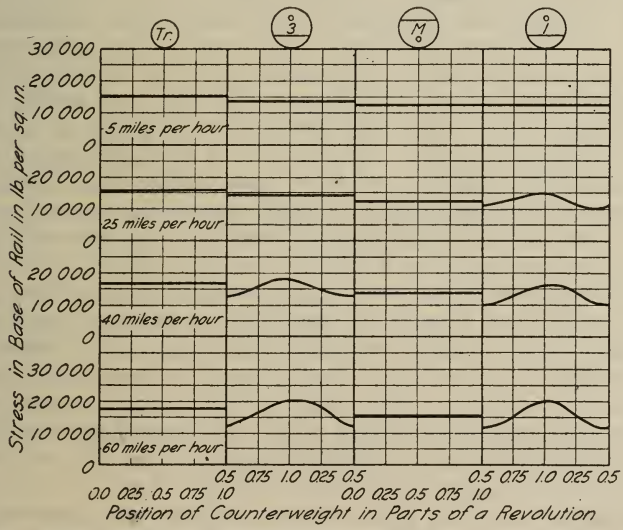


FIG. 32.—CURVES OF AVERAGE STRESS IN RAIL ON STRAIGHT TRACK THROUGHOUT REVOLUTION OF DRIVER—SERIES 5136-49, PACIFIC TYPE LOCOMOTIVE OF THE D. L. & W. R. R.

In Tables 10 to 18 are given the stresses with the counterbalance up and counterbalance down and the mean value, and also the calculated additional stress due to counterbalance at one speed, based upon the assumed values of counterbalance given in the previous tables without correction for the outside parts rotating in planes away from the driver.

It will be noted from the diagrams and tables that the highest stress in rail under the main driver in all the locomotives, except the Pacific type of the D. L. & W. R. R. and the Balanced Compound Prairie type of the A. T. & S. F. Ry., occur when the counterweight is up. For several of the locomotives a calculation for the counterbalance in the main driver shown in the tables would indicate that in many cases the high stress may be expected to occur when the counterweight is down, if the method of calculation used is that so usually employed of neglecting the effect of the plane of motion of the main rod and other outside rotating parts not being coincident with the plane of motion of the counterweight and wheel and rail, whereas the opposite condition may be shown by the results. It is evident from the discussion of this subject in the second progress report that a proper consideration of the relative positions of the plane of the rotating parts and of the wheel will account for this seeming contradiction.

Two locomotives of those used have characteristics which differ from those just described. The Pacific type locomotive of the D. L. & W. R. R. has no appreciable counterbalance effect under the main driver (see Fig. 32); this may be expected from the data of the counterbalance of Table 8. The Balanced Compound Prairie type locomotive of the A. T. & S. F. Ry. has four connecting rods; there is fair counterbalance when these are at the high and low position (see Fig. 24 and 31). When the counterweight is half way between high and low position there is an unbalanced effect; at 50 miles per hour the additional stress due to this is 6,200 lb. per sq. in. for one position and 3,400 lb. per sq. in. for the other. The increased stress in rail due to counterbalance in this locomotive is then greater than that found for the other locomotives except the Light Santa Fe type, and this notwithstanding it may have been expected that the balanced compound would give low counterbalance effects.

In the case of the Light Santa Fe type locomotive the smallness of the diameter of the wheel prevented the application of sufficient counterweight to the main driver to give a proper counterbalance, even though bobs were employed. For the Heavy Santa Fe type locomotive, the counterweight on the main driver was nearly as heavy as that required when all the rotating parts are considered to be in the same plane as the wheel, but the high stress for counterweight up shows the need for a revision of such calculations, as do the other locomotives.

A study of the values of the stresses attributable to counterbalance, as given in the tables and diagrams indicates that these streses are not unreasonably high with the exception of the Light Santa Fe type locomotive.

In the Pacific type locomotive of the A. T. & S. F. Ry. the additional stress in rail attributable to counterbalance at a speed of 60 miles per hour is 5,500 lb. per sq. in. under the third driver, which is 29 per cent of the average stress found at 5 miles per hour. The highest stress under the main driver occurs when the counterweight is up.

In the Light Prairie type locomotive the highest counterbalance stress at 50 miles per hour is under the first driver, 5,200 lb. per sq. in., which is 33 per cent of the average stress under this driver at 5 miles per hour.

In the Mountain type locomotive the highest counterbalance stress at 60 miles per hour is under the main driver when the counterweight is up, 2,900 lb. per sq. in., which is 21 per cent of the average stress under this driver at 5 miles per hour. It is well to note that this locomotive rode smoothly and evenly. The counterbalance stresses were unusually small.

The Light Santa Fe type locomotive (an old design, the first of this type) gives relatively high counterbalance effects. Although the driver loads are small, only about 83 per cent of the wheel loads of the other locomotives tested, the additional stress in rail under the first driver, due to counterbalance, is 7,700 lb. per sq. in., and under the main driver 9,200 lb. per sq. in. These counterbalance effects are 64 per cent and 113 per cent, respectively, of the average stresses under these drivers at 5 miles per hour, unusually large counterbalance effects. The observed stresses themselves are not exceptionally high, the rail being relatively heavy for the wheel load of this locomotive, but the stresses indicate very poor counterbalancing. This condition of counterbalance is doubtless due to inability to add adequate counterweight to such small drivers.

For the Heavy Santa Fe type locomotive the greatest increase in stress in rail due to counterbalance was under the fourth driver, 5,300 lb. per sq. in. at 40 miles per hour, which is 43 per cent of the average stress in rail under this driver at 5 miles per hour. The next greatest counterbalance stress was under the first driver. The greatest stress under the main driver occurred when the counterweight was up. This locomotive rode smoothly and easily even at 40 miles per hour.

For the Pacific type locomotive of the D. L. & W. R. R. the greatest increase in stress in rail due to counterbalance was 4,300 lb. per sq. in. under the first driver at 60 miles per hour, which is 35 per cent of the average stress in rail under this driver at 5 miles per hour. The third driver gave about as great counterbalance stress as this at 60 miles per hour. Under the main driver the counterbalance effect was so small as not to give a definite deviation from a straight line representation of the tests. This locomotive was in general a smooth riding locomotive.

The Mikado type locomotive of the D. L. & W. R. R. was not run faster than 25 miles per hour on straight track; at the three speeds used the plotted points showed no counterbalance effect.

In general for all the locomotives except the Light Santa Fe type the stresses attributable to counterbalance at the highest speeds run, which were in some cases greater than the scheduled maximum speed

allowed for the locomotive, are less than 5,000 lb. per sq. in., and in most cases considerably less than this. The counterbalance effect at the highest speeds used may be said generally to be not more than 30 to 40 per cent of the average stress in rail at 5 miles per hour; in some few cases it runs higher, but in most cases it is lower. A counterbalance effect of 30 per cent will not be considered to be excessive, and the only question is whether the scheduled speeds are likely to be exceeded very much. It is seen that an increase of speed of 25 per cent will increase the counterbalance effect by 56 per cent of itself and an increase of speed of 41 per cent will double the counterbalance effect. It is evident that great care must be exercised to prevent the scheduled limits of speed being exceeded very much if the stresses due to counterbalance are to be kept within moderate bounds. It is also clear that every care should be taken in the design, construction and maintenance of the proper counterbalance, and checks should be exercised to insure that the conditions of counterbalance are known with certainty, since it has frequently happened that the counterbalance effect has varied from the reported or expected condition. Attention should be given to the main driver and a careful determination made of the effect of the outside rotating parts not being in the plane of the wheel and counterweight.

In the locomotive for which the weights and positions of the rotating parts seem accurately known, the value of the additional stress due to counterbalance calculated by the use of the calculated vertical dynamic augment agrees fairly closely with the observed counterbalance stress as found by the method already described. For the main driver the difficulties are increased, but accurate knowledge of the position of the several outside rotating parts with reference to the plane of the driver itself should permit a fairly close determination of the counterbalance stress under this driver. Any method of design or improvement in materials that will help to keep the counterbalance effect small should be welcomed by both the mechanical and the engineering departments.

8. **Effect of Speed and Combined Effect of Speed and Counterbalance.**—If the stresses in rail found in tests made throughout the revolution of a driver be averaged, the effect of counterbalance may be considered to be eliminated. The ratio of this average of the stresses throughout the revolution for a given speed to the average stress at 5 miles per hour may be taken to represent the effect of speed alone. In general the tests show that the effect of speed upon stress in rail under the drivers in all except the Light Santa Fe type locomotive was not large. The speed effect under the leading truck wheels, trailer and tender wheels is also of interest. The average values of the stress in rail throughout the revolution of the drivers and of the other wheels may be found in the tables and diagrams already referred to.

In Table 19 the principal values of the percentages of increase in stress in rail under the various wheels of the locomotives, due to increase in speed above 5 miles per hour are recorded, together with values of the increase due to combined speed and counterbalance effect under the

drivers. The general values given represent increases that are common to several wheels, and the maximum value relates to the highest of the group of wheels.

With the exception of the Light Prairie type and the Light Santa Fe type it will be seen that the general increase in stress in rail under the drivers due to speed alone in changing from 5 miles per hour to the highest speed used in the tests ranges from 15 to 27 per cent, which may be called a moderate and allowable speed effect. With the same exceptions the maximum speed effect under any one driver ranges from 24 to 30 per cent. For these locomotives and these tracks, then, the increase in stress

TABLE 19

Principal Values of Increase in Stress in Rail at a Given Speed Over the Stress at 5 Miles Per Hour.

The increase is given in per cent of the stress at 5 miles per hour.

Type of Locomotive	Drivers				Trailer	Tender Wheels	
	Speed Alone		Combined Speed and Counterbalance				
	General Value	Maximum Under One Driver	General Value	Maximum Under One Driver		General Value	Maximum Under One Wheel
A. T. & S. F. Ry.							
Pacific 60 m. p. h....	27	30	53	53	15	75	93
Balanced Compound Prairie Type							
50 m. p. h.....	15	26	30	43	29	50	87
Mountain 60 m. p. h....	20, 20	30	35	50, 55	10	40, 60	110
Light Santa Fe							
45 m. p. h.....	40	77	105	190	10	25	35
Heavy Santa Fe							
40 m. p. h.....	18, 18, 18	28	50	80	15	75, 70, 50	110
D. L. & W. R. R.							
Pacific 60 m. p. h....	20	24	42	59	14	45	57

in rail due only to the change in speed noted may be said generally to be less than 35 per cent.

For the combined effect of speed and counterbalance for the same locomotives for the maximum speed used, the range of general total increase in stress in rail is 30 to 53 per cent, and the maximum increase under any driver ranges from 43 to 80 per cent. The stresses in rail at the maximum range from 20,000 to 29,000 lb. per sq. in the three sections of rails used. It should be borne in mind that these stresses are the averages of the stresses in the two edges of the base of rail, and also that they do not take into account the variations from the average which are bound to occur, nor the effect of speeds higher than those normally used. It should be remembered also that the track was substantial track in good condition and that the locomotives were of good design and in good working order.

The increase in stress due to speed alone for the Light Prairie type locomotive by the method of analysis used is very small, much smaller than for the other types of locomotive and varies from nothing to 5 per cent. No explanation for this is offered. The combined effect of speed and counterbalance (by the method of analysis used, most of this is effect of counterbalance) is 40 per cent for the first driver and about 20 per cent for the other two drivers.

Although the stresses found with the Light Santa Fe type locomotive at the maximum speed were not exceptionally high, the driver loads being light, and their spacing close, the percentage of increase in stress in rail under the drivers for both speed and combined speed and counterbalance was markedly high. The increase in stress due to speed alone was 47 per cent under the fourth driver and 77 per cent under the main driver; the increase in stress due to combined effect of speed and counterbalance under the several drivers was 87, 74, 190, 97 and 73 per cent, respectively. It may be added that this locomotive was built to be used with a rail lighter than the 90-lb. S. F. rail section that was used in the track tested.

In none of the locomotives tested was the effect of the counterbalance of the drivers transmitted to the trailer in sufficient amount to be brought out in the stress-position of counterweight curves, as was the case in some of the locomotives discussed in the second progress report. The increase in stress under the trailer, due to speed, was in general less than that under the drivers, though the value for the Balanced Compound Prairie type locomotive (27 per cent) was as great as under any driver. In general the increase noted at the highest speed was from 10 to 15 per cent.

For the locomotives with a two-wheel leading truck, the proportionate increase of stress in rail under the truck due to speed is generally somewhat less than that found under the drivers. The value of the increase in stress at the highest speed runs about 20 per cent. For the locomotives with four-wheel leading trucks, the increase under the front wheel is about 35 per cent for the two Pacific type locomotives and 20 per cent for the Mountain type. The increase under the rear wheel is much greater proportionally than that under the front wheel, though as the stress under the rear wheel at static load is very small, because of the proximity of the first driver, the stress at the highest speed is not large, generally less than that under the front wheel.

The effect of speed upon stresses in rail under the wheels of the tender varies with the type of locomotive; the percentage of increase is generally considerably larger than is found under the drivers. In the Pacific type of the A. T. & S. F. Ry. and the Heavy Santa Fe type, the increase of stress in rail at the highest speed is approximately 75 per cent. In the other locomotives the range is from 40 to 60 per cent. The stresses at both 5 miles per hour and at the maximum speed are smaller than the corresponding stresses under the drivers. How much of this large proportional increase in stress is due to the smallness of the wheel

diameter and how much to the relatively small load on the wheels or to the way in which the truck is directed and even to the condition of the springs are not known.

9. **General Discussion of Tests on Straight Track.**—With the exception of those found with the Light Santa Fe type locomotive the tests indicate no abnormal stresses in rail under any of the wheels at any speed. The locomotive design is generally within the requirements of what has been considered acceptable practice by mechanical departments. With a single exception the locomotives rode smoothly, showing that there was probably no abnormal strain on the frame of the locomotive. As there are differences in the division of the loads and in the counterbalancing and in the spacing of wheels, some consideration and comparison of the stresses developed in rail under the various wheels of the locomotives may be useful.

In most cases the loads on the drivers are moderately heavy, in the neighborhood of 30,000 lb. on a driver, reaching 33,000 lb. in the case of the Pacific type locomotive of the D. L. & W. R. R. The load on the trailers differs considerably. The leading truck wheels, either two-wheel or four-wheel trucks, each carries about half as much load as a driver. Part of the tenders have four-wheel trucks and part six-wheel; the wheel loads of the latter are about the same as those of the former.

The spacing of the drivers is, of course, greater for the Pacific and Prairie types than for the others; the greater spacing results in a somewhat greater proportional effect of the weight upon the stresses developed under the drivers. On the other hand the shorter spacing of the drivers in the Light Santa Fe type locomotive decreases the stresses.

Leaving out of consideration the Light Santa Fe type locomotive of the A. T. & S. F. Ry. and the Pacific type locomotive of the D. L. & W. R. R., the latter having heavier driver loads, it is seen that the several locomotives develop stresses under the drivers at 5 miles per hour that do not differ greatly if allowance is made for differences in rail section. Also there is generally no great difference in stresses for any group of drivers, the stress under the first and last driver naturally being somewhat greater than under the others, as is indicated in Fig. 6 of the first progress report. In the Heavy Santa Fe type locomotive, the stresses vary more, ranging from 10,100 to 15,900 lb. per sq. in. The latter stress, which is under the fifth driver, is even greater than would be expected from analytical considerations (the stress under the trailer being correspondingly less), indicating that the division of load is not that given in the diagram of locomotive loading.

In general the effect of speed alone (omitting the effect of counterbalance) is as small as could be expected. Even the 27 per cent increase at 60 miles per hour found with the Pacific type locomotive of the A. T. & S. F. Ry. seems moderate. The increase of 18 per cent at 40 miles per hour found with the Heavy Santa Fe type locomotive, compared with the Pacific type locomotive, is directly proportional to the increase of speed,

so that this freight locomotive appears to produce about the same relative increase as the passenger locomotive, if the conditions of the truck are considered to be the same. It will be seen that the increase at the drivers that show the maximum speed effect gives somewhat the same comparison, though the increase for one driver of the Santa Fe type (28 per cent) is relatively larger.

The percentage of increase in stress in rail due to counterbalance at the highest speeds used with the different locomotives (see Table 19) generally shows an average for the several drivers that is about the same as the general value for the increase in speed alone. In the case of the Heavy Santa Fe type locomotive the increase due to counterbalance is greater proportionally than for the other types; this may be due to a poorer opportunity for counterbalancing. The maximum increase due to counterbalance for some one driver of a locomotive is generally considerably greater than the average for the drivers; this shows a need for more careful study of the counterbalancing problem. Further, attention should be called to the discrepancy between the condition of counterbalance in the main driver usually assumed in design and the condition found in the test. This discrepancy was found in every locomotive tested; the difference is explainable at least partially by the neglect to recognize the effect produced by the outside rotating parts not being in the same plane as that of the wheel and rail. It would appear also that the counterbalancing of other drivers of some of the locomotives might be improved. It should be noted also that for speeds still higher than those used the effect of counterbalance may be expected to increase as the square of the velocity.

The general values given in Table 19 for the general increase in stress at the maximum speed used due to combined speed and counterbalance (excepting for the Light Santa Fe type as before) range from 30 to 53 per cent. It would appear that these values may be considered to be representative of the increase which may be expected with locomotives of fair design in good condition and with track in good condition. At an individual driver on the Heavy Santa Fe type locomotive the increase runs up to 80 per cent. If speeds still higher than those used in the tests are likely to be run even occasionally, the fact that the stress in rail will be increased more rapidly than as the first power of the speeds should receive consideration in judging of the effect on rail and track.

With the exception of the Light Santa Fe type locomotive, the nominal load on the trailer ranges from 72 to 109 per cent of the average load on the drivers. It will be recalled that analytical considerations show that with the trailer well away from the rear driver the stress in rail will be greater accordingly than that under a driver, and that to give an equal stress under static conditions the load on the trailer should be in the neighborhood of 75 to 80 per cent of that on a driver. (For an illustration of this effect see Fig. 6 of first progress report.) The observed stresses bear out the relations shown by the analysis except in the case of the Mountain type and the Heavy Santa Fe type locomotive, where

the observed stresses under the trailer at 5 miles per hour were 15 to 20 per cent lower than the calculated stresses and the division of load among drivers and trailer was evidently not as reported on the locomotive diagrams, the higher stresses under the drivers indicating that the drivers were taking a greater share of the load than was planned and the trailer less. The stresses under the trailer of the locomotives of the D. L. & W. R. R. ageed closely with the calculated values. In four of the types of locomotives used in the tests, the stress under the trailer at 5 miles per hour is greater than the average stress under the drivers. The excess of stress under the trailer over the average stress under the drivers at a speed of 5 miles per hour was as follows: Pacific type, 15 per cent; Mountain type, 22 per cent; Heavy Santa Fe type, 31 per cent, and Pacific type of the D. L. & W. R. R., 18 per cent. The increase in stress under the trailer due to speed is generally less than the increase in stress under the drivers due to speed alone, and when the effect of counterbalance is considered much or all of the discrepancy between the stresses under the trailer and the drivers may disappear, and there will also be the lesser effect of the depression of the track by a single load. It should be recalled that in the tests recorded in the second progress report there was little difference in effect of speed between trailer and drivers.

For the purpose of comparing the effect of two-wheel and four-wheel leading trucks, the Pacific and Mountain type locomotives, both having four-wheel trucks, may be compared with the Prairie and the Heavy Santa Fe type locomotives, having two-wheel trucks. One wheel of the four-wheel truck carries about one-half as much load as a driver, and in the two-wheel truck a wheel also carries about one-half as much as a driver. In the four-wheel truck the leading wheel develops 50 to 60 per cent as much stress in the rail at 5 miles per hour as does a driver; a second wheel produces 25 to 40 per cent as much as a driver, the smaller values under this wheel being due to the proximity of the leading driver. With the two-wheel truck the stress in rail under the wheel is 50 to 60 per cent of the stress under a driver. The highest stresses under a truck wheel are found under the leading wheel of the Pacific type locomotive (13,800 lb. per sq. in. at 60 miles per hour) and under the wheel of the Balanced Company Prairie type locomotive (14,800 lb. per sq. in. at 50 miles per hour). These are not unduly high stresses. Those found with the other types are smaller, but all of them are great enough to insure sufficient pressure on the rail and sufficient depression of rail relatively to that under the other wheels to give stable conditions. It would appear that so far as straight track is concerned either two-wheel or four-wheel trucks will give stresses in rail which should be considered acceptable; this conclusion allows latitude in the designing of the locomotive.

The following locomotives have six-wheel trucks under the tender: Pacific, Mountain and Heavy Santa Fe types of the A. T. & S. F. Ry. All the other locomotives tested have four-wheel tender trucks. The weights on the individual wheel when the tender is fully loaded do not

differ greatly, ranging from 19,000 to 23,500 lb., except in the Light Prairie type. At the time of test the effort was made not to have the loads vary much, water being taken frequently; but the usual load was less than full load. The stresses in rail on straight track at a speed of 5 miles per hour were generally from 8,000 to 10,000 lb. per sq. in., being in the neighborhood of two-thirds of those under the drivers. Although, as already noted, the average increase at the highest speed is 40 to 75 per cent, the resulting stress at the highest speed is usually considerably less than the average stress developed under the drivers, though in the Heavy Santa Fe type locomotive the stress under the wheels of the tender at the highest speed reached almost as great a value as the average stress under the drivers (about 16,000 lb. per sq. in.). The great variation in the effect of speed under the wheels of the tenders of the various locomotives agrees with the results noted in the second progress report. It seems that there must be considerable difference in the design and construction of tenders or in their maintenance.

The placing of a double trailer (two-trailer axles) under a Heavy Santa Fe type locomotive for the purpose of learning experimentally the effect of this form of construction in changing the stresses in rail under the trailers and under the drivers was a novel undertaking that gave results of much interest. Although the double trailer was put in one of the regular locomotives and thus the design did not have all the proportions which would be desired in a new design, it gave very successful results and decreased the stresses under both drivers and trailer, at the same time working smoothly and satisfactorily. The introduction of the double trailer brings a marked change in the distribution of stress under the several wheels of the locomotive. The stress under the trailer in the 90-lb. S. F. rail is changed from 16,900 lb. per sq. in. in Locomotive 3813 to 9,000 lb. per sq. in. under each trailer in Locomotive 3829. Considerable load has evidently been removed from the last three drivers, for the stress at 5 miles per hour was decreased 40 per cent under the fifth driver. With these changes have come an increase of 60 per cent in the stress under the wheel of the front truck, though the resulting stress is not especially high on the straight track. The speed effect under the trailers is 25 per cent increase of stress in the change from 5 to 40 miles per hour. Even at 40 miles per hour the stress in the 90-lb. S. F. rail under the trailers is only 11,300 lb. per sq. in. and the stresses under the drivers (not including the effect of counterbalance) are only about 13,700 lb. per sq. in. The stresses indicate a smoothly running locomotive on straight track. It turns out, however, that the presence of two-trailer axles does not give adequate room or load to permit the addition of booster equipment, which is now desired with this type of locomotive.

Another locomotive not included in the preceding discussion is the Light Santa Fe type. It differs from most of the other locomotives in having lighter driver loads and in having nominally a very light load on the trailer. The spacing of the drivers is somewhat closer and the trailer

is but 7 feet away from the fifth driver. All of these differences favor the development of small stress in the rail. At 5 miles per hour the stress under the first driver is more than 50 per cent greater than that under the third, fourth and fifth drivers. The stress under the trailer is double the value calculated from the weight given on the locomotive diagram. It is evident that the weights are not distributed as given in the diagram; the average of the observed stresses under drivers and trailer, however, closely agrees with the average of the calculated values. This locomotive is marked by large increase in stress in rail under the drivers, due to both speed and counterbalance. A change in speed from 5 to 45 miles per hour increases the stress in rail under the main driver 77 per cent for speed effect alone, and 113 per cent for counterbalance alone, the combined effect of speed and counterbalance being an increase in stress of 190 per cent. Due to the lighter loads and close spacing of the wheels the stresses in rail even at 35 miles per hour are below 20,000 lb. per sq. in. in the 90-lb. S. F. rail. At 45 miles per hour higher stress is found. This is a rough riding locomotive. Its behavior in causing lateral bending of the rail, which is discussed farther on, and the high increases in stress with increased speed are undesirable features, though it should be noted that the first of these locomotives were built in 1902 and were the first of the 2-10-2 class built by any road.

The calculation of stresses in rail by the analytical method given in the first progress report in general gives results which agree closely with the observed stresses at 5 miles per hour. The calculated stresses are given in Tables 10 to 18. In some cases the stress under one wheel is higher and that under another is lower, but the average of the stresses under all the wheels checks closely, indicating slight differences between the actual and the nominal distribution of load among the drivers. In one or two cases the distribution of load evidently differs from the nominal values. Some change in distribution is sometimes seen when the speed is increased. This agreement of values should give added confidence in the accuracy of the analytical method.

10. **Stresses at the Two Sides of the Locomotive.**—In view of the experience with one of the locomotives reported in the second progress report, it was thought desirable to make a study to learn whether the stresses in the rail on one side of the locomotive differ from those on the other side. Tests with three locomotives were made on straight track with four instruments on the south rail and four instruments directly opposite on the north rail. The first day the locomotive was run eastward at the several speeds, and the second day its direction was reversed and runs were made over the same track at the same speeds. The instruments were left on the rail over night and other conditions remained the same. The four instruments on the south rail of the eastward runs were averaged with the four instruments of the north rail of the westward runs, giving the stresses for the right side of the locomotive. Similarly the instruments at the left side of the locomotives were averaged for the

two directions. It would seem that the tests obviated possible differences due to variations in instruments and variations in conditions of the track on the two sides, and that, therefore, the results of the tests for the two sides of a locomotive may properly be compared.

In Fig. 33 the stresses under the sides of the Mountain type locomotive are given. It is seen that the stresses under the left side are nearly the same as those under the right side. The only variation of any amount is under the trailer at 5 miles per hour.

In Fig. 35 the stresses under the two sides of the Mikado type locomotive of the D. L. & W. R. R. are given for speeds of 5, 15 and 25

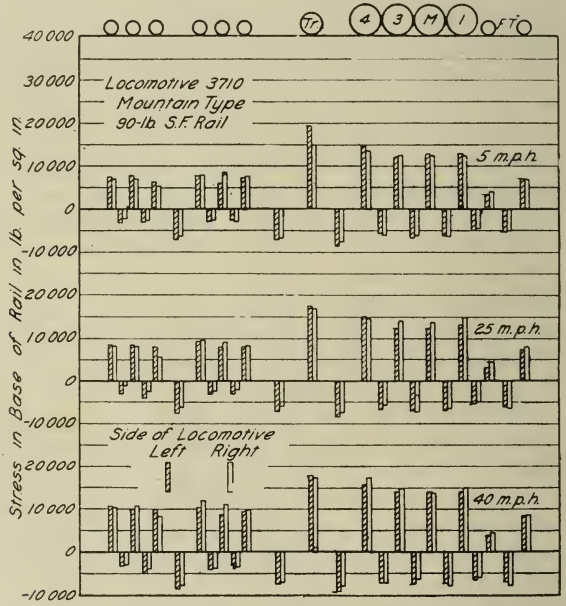


Fig. 33.—Mean Stress in Rail at the Right and Left Sides of the Mountain Type Locomotive of the A. T. & S. F. Ry.

miles per hour. At 5 miles per hour the stresses under the two sides of the locomotive are very nearly equal. At speeds of 15 and 25 miles per hour the stresses under the left drivers are somewhat more than those under the right drivers, the reverse being true for the trailer. The stessses under the left side of the tender at the three speeds average about 10 per cent. higher than those under the right side. However, none of these average stresses is as high as 15,000 lb. per sq. in.

Fig. 34 presents the results under the two sides of the Pacific type locomotive of the D. L. & W. R. R. at speeds of 5, 25, 40 and 60 miles per hour. At the speeds of 5, 25 and 60 miles per hour the stresses under the

two sides of the locomotive and tender average approximately the same, but at the speed of 40 miles per hour the stresses under the left side averaged about 10 per cent more than under the right side. None of the average stresses is greater than 18,000 lb. per sq. in.; in no case does the stress under the right and left wheel of an axle differ more than 2,000 lb. per sq. in. and generally the difference is much less than this.

It may be said then that for both types of locomotive the stresses

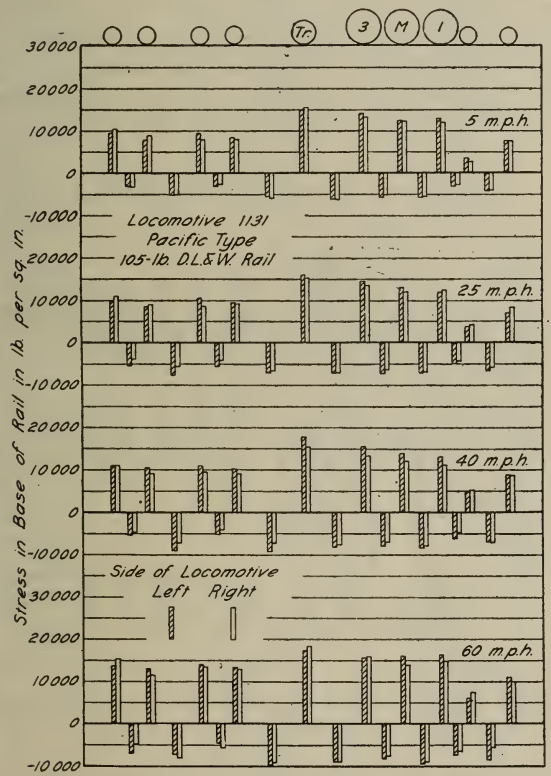


Fig. 34.—Mean Stress in Rail at the Right and Left Sides of the Pacific Type Locomotive of the D. L. & W. R. R.

in rail under the two sides of the locomotive were found to differ but slightly. It appears that it more frequently happens that the stress under the left wheel of an axle is somewhat greater than that under the right wheel than does the reverse; about two times out of three the stress at the left side is greater than at the right. There seems to be no noticeable difference in this relation at the lower and the higher speeds, so that the

variation found with these engines is not caused by speed or counterbalance. Besides, the tender wheels show the same general variation as the wheels of the locomotive. It seems probable then that in these locomotives the variations between the stress in the rail at the left and the right sides are no more than are to be expected with any well-built machine.

11. **Stresses at the Two Edges of the Base of Rail.**—In the two preceding progress reports reference was made to the frequent and common lateral bending of the rail in straight track and it was shown that the stresses due to such bending attain considerable magnitude. The data of the tests described in this report give added information on the subject.

In Fig. 36 is plotted the ratio between the stress at the outside edge of rail and the average of the stresses at the two edges (mean stress in base of rail) for each observation under the first pair of drivers of the

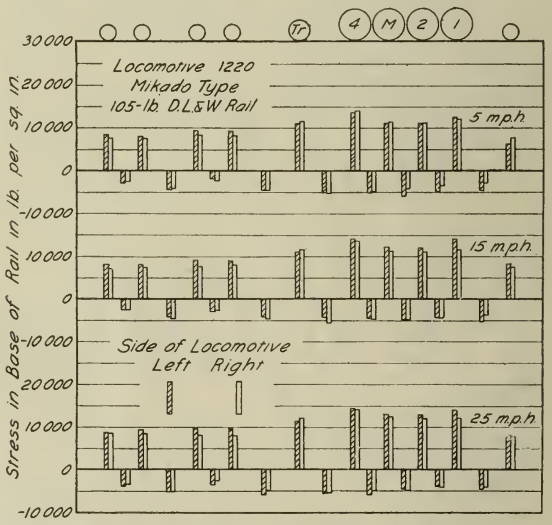


Fig. 35.—Mean Stress in Rail at the Right and Left Sides of the Mikado Type Locomotive of the D. L. & W. R. R.

Light Santa Fe type locomotive and at three speeds. The ratio of the stress at the inside edge of rail to the mean stress in base of rail may be found by subtracting the ratio given in the diagrams from 2. It should be noted also that the ratio of stress at outside edge to that at inside edge may be found by dividing one of these ratios by the other. The value of the stress which corresponds to an observation that gives a ratio of 1.00 is the same as the average stress for the given wheel as it is recorded in the diagrams and tables of general stress values. A mean line drawn through the average of the ratios is also shown. Fig. 36 also gives the stresses at the inside edge and at the outside edge for the same drivers.

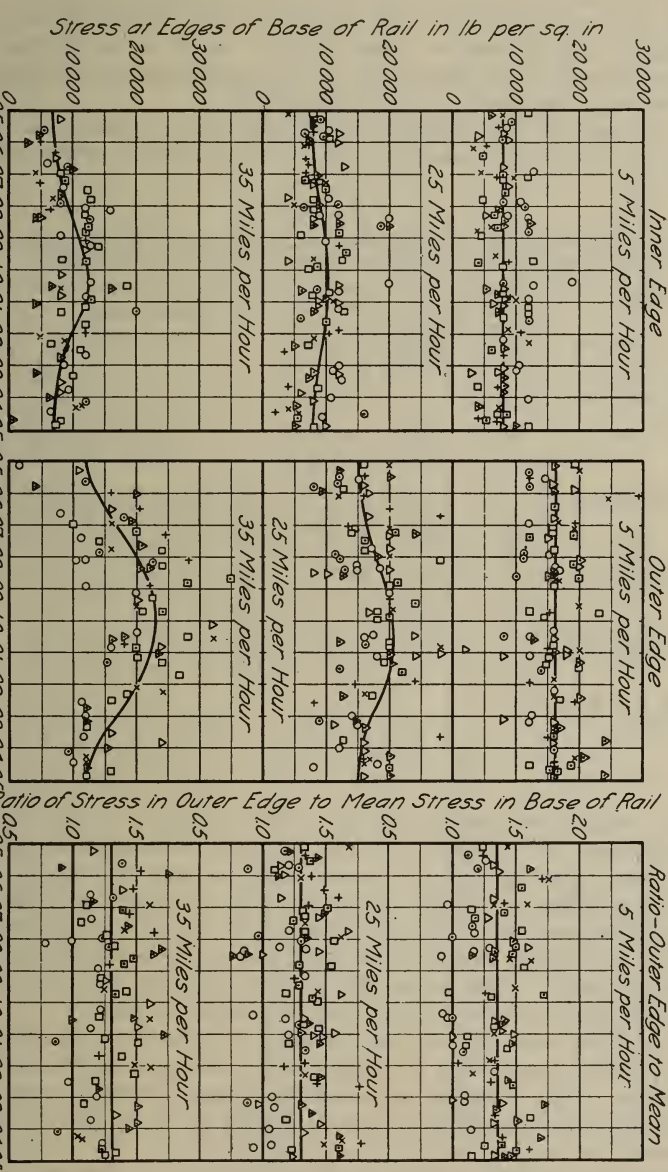

Stress at Edges of Base of Rail in lb per sq. in
Inner Edge
Outer Edge
5 Miles per Hour
25 Miles per Hour
35 Miles per Hour
Position of Counterweight in Parts of a Revolution
Ratio of Stress in Outer Edge to Mean Stress in Base of Rail
Ratio-Outer Edge to Mean

Fig. 36.—Stresses at Inner Edge, Outer Edge, and Ratio of Stress in Outer Edge to Mean Stress in Base of Rail Throughout the Revolution of the First Driver of the Light Santa Fe Type Locomotive on Straight Track.

Fig. 37 gives similar data for the first pair of drivers of the Heavy Santa Fe type locomotive. Table 20 gives values of the average ratio of the stress at outer edge of rail to the average stress at the two edges for some of the wheels of several locomotives.

A study of the diagrams (confirmed by an examination of the diagrams of the other wheels, which are not reproduced) shows consider-

TABLE 20

General Values of the Average Ratio of the Stress at Outer Edge to Mean Stress in Base of Rail for Wheels of Several Locomotives.

Locomotive	Speed in m. p. h.	Average of Tender Wheels	Trailer	Driver Number					Front Truck Wheels
				5	4	3	2	1	
A. T. & S. F. Ry.									
Pacific	5	1.08	1.05	.....	.....	1.17	1.14	1.12	1.08 1.01
	25	1.06	1.04	.....	.....	1.14	1.12	1.12	0.96 1.05
	40	1.03	1.07	.....	.....	1.08	1.10	1.12	1.05 1.05
	60	0.99	1.07	.....	.....	1.08	1.13	1.17	1.08 1.01
Light Santa Fe	5	1.00	0.96	1.10	1.10	1.01	1.26	1.33	1.05
	25	1.01	0.97	0.99	1.06	1.05	1.29	1.31	1.04
	35	1.00	0.99	0.96	1.09	1.05	1.24	1.31	1.00
	45	1.04	0.94	1.00	1.02	1.07	1.27	1.20	1.04
Heavy Santa Fe	5	1.13	1.03	1.08	1.18	0.97	1.16	1.12	1.01
	25	1.07	1.01	1.05	1.15	0.93	1.10	1.06	0.97
	40	1.00	1.05	1.03	1.13	0.89	1.10	1.06	0.97
Double Trailer Heavy Santa Fe	5	1.03	1.02 0.98	0.96	0.98	0.97	1.00	1.05	0.97
	25	1.05	1.04 1.01	0.97	0.98	0.94	1.01	1.01	0.96
	40	1.05	1.05 1.05	1.08	1.05	0.99	1.05	1.07	1.04
D. L. & W. R. R.									
Mikado	5	1.06	1.00	.....	1.06	1.14	1.24	1.17	1.20
	15	1.05	0.96	.....	1.04	1.18	1.31	1.15	1.14
	25	1.04	1.01	.....	1.07	1.16	1.23	1.13	1.13

able variation between the stresses at the two edges. The stress at the outer edge of the base of rail is quite frequently 33 per cent or more greater than the mean stress in base of rail, in a few cases reaching an excess of 50 per cent or more, and this holds true throughout the revolution of the driver. A stress at the inner edge 33 per cent greater than the mean stress and even more was found, though less frequently. This means that in these cases the stress at one edge of the base is twice as much as that at the other, or more.

Another way of considering the range in stress at the two sides of the rail, which seems useful in judging of the amount and effect of lateral stresses, is by noting for any wheel the range of stress on each side of the mean line in a diagram such as Fig. 36 or 37. It will be seen that usually the stresses are fairly well distributed in a belt between limits on the two sides of a mean line; these limits may be said generally to be at a distance above and below the mean line of 4,000 or 5,000 lb. per sq. in. of stress, for the stresses and rail sections used in the tests. It may be said then that the average value of the ratio between the stress at the

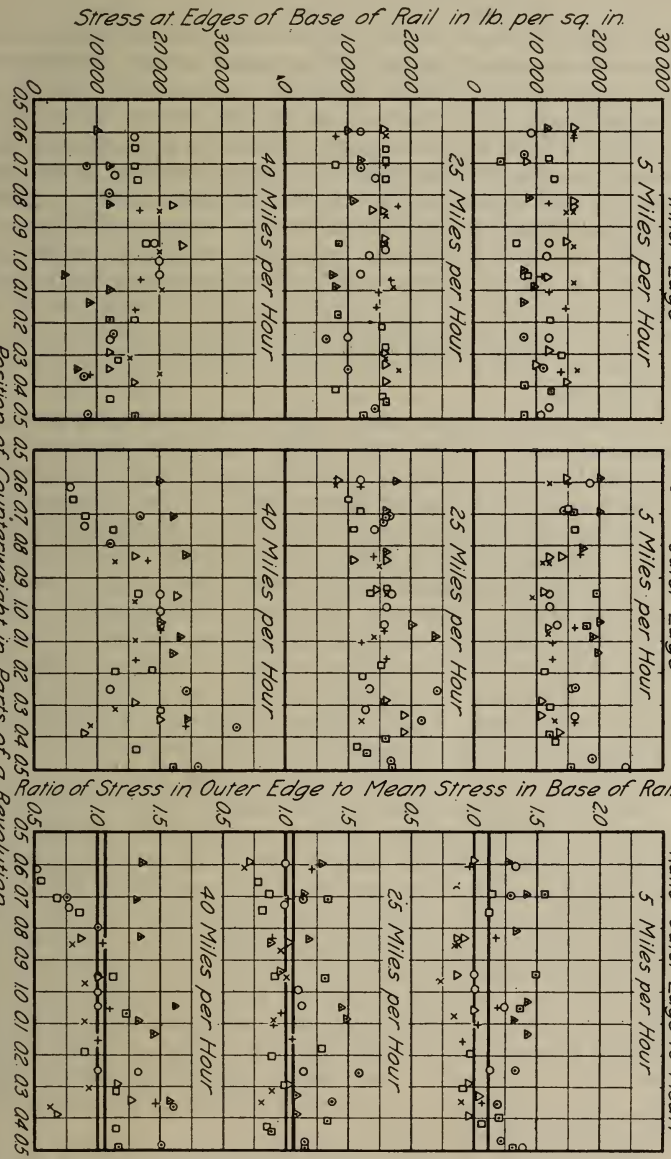

Stress at Edges of Base of Rail in lb. per sq. in.
Inner Edge
Outer Edge
5 Miles per Hour
25 Miles per Hour
40 Miles per Hour
Position of Counterweight in Parts of a Revolution
Ratio of Stress in Outer Edge to Mean Stress in Base of Rail
Ratio-Outer Edge to Mean

Fig. 37.—Stresses at Inner Edge, Outer Edge, and Ratio of Stress in Outer Edge to Mean Stress in Base of Rail Throughout the Revolution of the First Driver of the Heavy Santa Fe Type Locomotive on Straight Track.

outside edge and the mean stress in base of rail under any wheel may be taken to represent the general effect of lateral bending under that wheel. At the same time the further fact must be considered, that the stress at the given edge of the base of rail may be expected to vary from a value 4,000 or 5,000 lb. per sq. in. more than this mean stress to a value as much less, this variation being due to the variations in the way the locomotive applies itself on different runs and being similar to variations expected in stresses resulting from bending in the vertical plane. We may then deal with the average ratio and average stress at the outside edge given by the observations, with the understanding that the stresses at the two edges may vary 4,000 or 5,000 lb. per sq. in. above and below the mean value for the rail sections and loads used in the tests, and that these variations may cause a change in amount or direction of the lateral bending of the rail.

In general this average value of the ratio between the stress at the outside edge and the mean stress in base of rail was somewhat greater than unity, though in certain of the wheels it was slightly less than unity. As shown in Table 20, for the Heavy Santa Fe type (locomotive 3813) the ratio under the first driver was 1.14, under the second driver 1.12, under the third (the blind driver) 0.93, under the fourth 1.15, under the fifth 1.05, and under the trailer and tender wheels about 1.05, as averages for the several speeds. Locomotive 3829 gives ratios nearer to unity. It appears that in locomotives of the same class the condition of the tires or other parts has much to do with the amount of the lateral bending of the rail. It is also apparent that different wheels of a locomotive produce lateral bending stresses of quite different magnitudes. At the blind driver there seems usually to be less outward bending stress than at other drivers. The ratio for the trailer is but little greater than 1.0.

The first and second drivers of the Light Santa Fe type locomotive present an illustration of very marked outward lateral bending of rail, the ratio of stress at outside edge of base of rail to mean stress at the first driver being 1.33 at the speed of 5 miles per hour. The ratios for the main driver and the trailer do not vary greatly from 1.00, being 1.01 and 0.96, respectively, at the speed of 5 miles per hour, indicating about as much lateral bending inwardly as outwardly. The average ratio under the first driver is markedly greater for the left rail than for the right. Whether this is due to differences in the condition of the tread of the two drivers on the same axle, to the difference in time of stroke of the two pistons, or to other causes is not known. There is no marked difference in the amount and nature of the lateral bending of the rails at the two sides of the other locomotives.

Since the lateral bending strength of the rail is less than the vertical bending strength, the bending moment corresponding to a given fiber stress is less for lateral bending than for vertical bending. The section modulus $\frac{I}{c}$ for bending in a vertical plane for the rail sections used is about five

times the section modulus for bending in a horizontal plane. For a ratio of stress at outside edge to mean stress with a value of 1.33, it is seen that the lateral bending moment in rail is about 7 per cent of the vertical bending moment required to develop the mean stress in the base of rail. A value of the lateral bending moment found quite frequently is 4 per cent of the vertical bending moment, though in a few cases it reached 14 per cent if the effect of the added 5,000 lb. per sq. in. of the belt of stresses be taken into consideration. From a study of the observed values it would seem that in designing a rail section for straight track a lateral bending moment at least equal to 14 per cent of the vertical bending moment should be provided for.

III. TESTS ON CURVED TRACK

12. **The Action of Curved Track.**—In traversing a curve the action of a locomotive or car upon the rail and track differs in several respects from the action on straight track. Some of the differences are of importance in the development of stresses in the rail and in the effect on the track structure as a whole. The principal elements of the action of locomotives and cars on curves that need to be considered in the study of the action of the curved track may be outlined as follows:

1. Since the outer rail is longer than the inner one, the wheel on the outer rail must slip backward or the wheel on the inner rail slip forward, or both, to overcome the difference in travel; this motion of itself may be expected to be accompanied by a force which acts in a longitudinal direction along the rail rather than laterally, and thus does not produce bending in the rail, though it may modify the magnitude of the force required to produce lateral slipping and hence indirectly affect the magnitude of the lateral bending force.

2. Since the driving wheels of a locomotive, and the wheels of a truck having more than one axle, are grouped in a frame of some stiffness, a lateral movement is involved in changing the direction of the motion of the group of wheels in traversing a curve, and this involves a slipping of one or more pairs of the forward wheels in a lateral direction inwardly of the curve and generally of one or more pairs of the rear wheels in a lateral direction outwardly. This action develops lateral pressures against both the inner and the outer rail. A lateral force is thus present at one or more wheels even when the speed is that corresponding to the superelevation of the track. The lateral pressure thus devleoped may be of considerable magnitude.

3. The lateral pressure at two or more of the drivers has been found to give a large increase in the stress at one edge of the base of rail. By reason of the lateral inclination of the track and the centrifugal force developed at the given speed, the division of load between the two rails will not be the same on curved track as on straight track, except at the one speed corresponding to the speed of superelevation; and even at that

speed the load will generally not be divided equally between the two wheels of most of the axles. It will appear also that there are transfers of load from one rail to the other that are due to other causes.

4. For some reason the action of the system of equalizing levers and springs of the locomotive is not the same on curves as on straight track; at least a much greater vertical bending stress is found in the inner rail under one of the intermediate drivers than under the others, especially at low speeds, and the abnormal action of the equalizers and springs may be one explanation for this. It is also true that a greater vertical bending stress is frequently found in the outer rail under the front driver than under others, the amount of the difference in both cases varying with the speed. This excess of stress under some of the drivers must be due to a large increase of load carried by these drivers.

5. An effect of the lateral inclination of the track and the presence of centrifugal force is the development of a lateral pressure against the rails; the amount of this lateral pressure may not be equally divided between the two rails or even among the several drivers on one side of the locomotive.

6. For sharp curves (curves of relatively small radius) there may be a spreading action on the track caused by the flanges bearing against the inner and outer rails; the lower limit of degree of curve above which this spreading action may be expected to become important will depend upon the length of wheel base and upon the difference between the wheel gage and the gage of the track. For the curves in ordinary use other sources of lateral pressure against the rail appear to be much more important.

The several elements of the problem just noted will now be considered somewhat more fully.

(a) Difference of Length of Outer and Inner Rail.—It is shown by Wellington* that the longitudinal slipping of the wheels of a freight car truck necessary to overcome the difference in length of the outer and the inner rail of a curve in some cases may occur on the outer rail and in others on the inner rail, though usually not by both wheels of an axle at the same time, since that wheel will slip which will slip more easily and a slight variation in either the load or the coefficient of friction will give one wheel or the other an advantage in this respect. Wellington further states that either the superelevation or the centrifugal force is alone competent to produce enough inequality of load to effect a difference and that one wheel having begun to slip is likely to continue so to do for some time, although it may not be the same wheel for two consecutive axles and change from one to the other wheel may take place from time to time. As most of the drivers of a locomotive will be found to be slipping in a lateral direction continually, the matter of the difference between the coefficient of friction for rest and motion will not enter into the question of whether the wheel will have longitudinal slip to the extent that might

*The Economic Theory of Railway Location, p. 285.

otherwise be expected from Wellington's analysis for freight car trucks. The fact that the drivers are coupled together by the side rods is an added feature which may make the action of locomotive drivers differ from that of car trucks. Wellington also shows that the coning of the wheels does not overcome the difference in the length of the rails, even for the curves corresponding to the coning, because the axle usually does not maintain a radial position on the curve, especially when grouped with others in a frame or truck; the coning, therefore, may be neglected as an element in the problem. The slipping of the wheel longitudinally on the rail results in a force being developed in the direction of the rail length (pull or push), which adds to the tractive pull required and produces wear of rail and wheel. It does not, however, cause bending stresses in the rail and so need not here be taken into account except as it may affect the way in which lateral slip on the rail occurs in combination with it and also as it may influence the magnitude of the lateral force, which is one component of the force required to produce the resulting slip on the rail.

(b) Changing the Direction of the Locomotive in Passing Around the Curve.—If in traversing a curve the axles of each pair of wheels took a radial direction (normal to the curve), the principal force required to change the direction of the motion of the truck or frame would be equal to that commonly called the centrifugal force; the centrifugal force and the transverse inclination of the track cause a lateral reaction against the rail. For a radial position of the axle the force producing lateral bending in the rail would then be the resultant of the centrifugal force and the component of the load due to the transverse inclination of the track.

When several axles are connected to a single stiff frame, as in a locomotive, quite different conditions are set up. The axles of the drivers must remain parallel to each other. The tendency of the whole group of drivers to go straight ahead must be overcome by a lateral force exerted by the outer rail against the flanges of one or more of the wheels at the front of the frame. In the case of the locomotive this turning movement is effected at the outer front driver, or at the outer wheels of the front truck, or by a combination of these truck wheels and the first driver. What wheel or combination of wheels will participate in this turning action may be expected to depend upon the curvature of the track, the design of the locomotive, and the stiffness and play of the front truck connection with the locomotive frame, and also in some cases upon the flexibility of the connections of the front pair of drivers with the other drivers of the locomotive. The action of these forward outer wheels then is to press against the outer rail, the outward lateral force thus exerted tending to increase the curvature of the outer rail; in some cases by reason of friction on the rail the truck wheel on the inner rail may exert a lateral pull in the same direction on that rail. As a result of this turning action, the front drivers (both inner and outer) will be made to slip laterally on the rail; that is, inwardly radially to the curve. Further back along the frame of the locomotive will be a point which will act as a center of rotation and about which the turning of the locomotive will take place.

If the lateral slip at any wheel is inward with reference to the curve (that is, toward the center of the curve), the effect on the rail is that of a lateral force acting inwardly of the curve, thus putting an inward lateral pressure on the inner rail and even on the outer rail, since the force may be transmitted by a wheel to the rail by friction at the top of the rail. The foregoing statement does not apply, of course, to those outer wheels whose flanges are active in causing the locomotive to change direction; for these the pressure against the rail is outward, tending to increase the curvature of the rail.

The lateral slip of the drivers in front of the point about which rotation takes place will be inward with reference to the curve; that of the drivers back of the point will be outward. It is apparent that the position of this center of rotation will depend upon the number of wheels in the frame, the position the wheels take with respect to the two rails, and the direction of the resultant slip of the wheels. If, as may happen, the axle of the next to the last pair of drivers remains normal to the curve, the center of rotation will be on the inner rail at this driver, provided, as

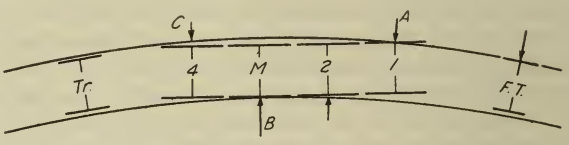


Fig. 38.—Position of the Wheels of a Mikado Type Locomotive When Traversing a Curve at Low Speeds.

is probable, the longitudinal slip of the drivers of this axle is all taken by the outer driver. It seems probable that the center of rotation in some cases may be somewhat ahead of the next to the last driver. It is true, of course, that the several drivers on one side may not lie exactly in a straight line; the lateral play of the journals will allow side movement, the amount depending upon the amount of the wear at the time. Fig. 38 represents the position of the wheels and the direction of a part of the lateral forces exerted by the rails on the wheels of the Mikado type locomotive at low speeds under conditions similar to those named. It is seen that the point of rotation is at or close to the third driver axle. The greatest lateral slip for any driver may be expected to occur under the front drivers. The lateral slip of the several drivers will then be proportional to the distance along the inner rail from the point of rotation to the inner driver of a given pair.

Generally the two-wheel front truck is connected to a radius bar, a swinging arm that is pivoted at a point back of the truck, and a limit of rotation is provided, so that after this limit is reached the flange of the outer wheels will press against the outer rail and thus aid in the turning action. To what extent the axles of the front truck will approximate a

radial condition on the curve and thus will have minimum lateral slip will depend upon the manner of the attachment and the sharpness of the curve. The trailing wheels are also pivoted on a swinging arm so that the axle possibly may take a radial position. When the limit of the swing of this arm is reached, these wheels also contribute to the stiffness of the frame and must be slid outwardly; thus an outward lateral force on the rails is developed at the trailers.

In the figures of Article 15, Lateral Bending of Rail on Curves, the position taken by the drivers and other wheels on curved track is shown for several locomotives. A study of these diagrams and of the amount and direction of the bending of the rails will indicate the position of the center of rotation of the locomotive. The relative amounts of the lateral slip at the several wheels may also be found rather closely.

It should be noted further that since both longitudinal and lateral slip on the rail will usually occur at the same time, resulting in slip in a diagonal direction, the force producing this slip of wheel in the diagonal

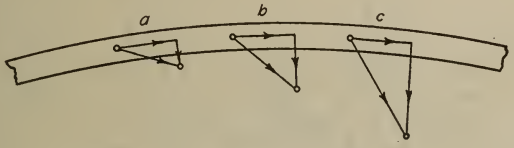


Fig. 39.—Longitudinal and Lateral and Resulting Slip of Wheels on the Rail.

direction (its magnitude being the force necessary to overcome friction) may be resolved into components along the rail and across the rail. In Fig. 39 the diagonal line represents relatively the direction and amount of the resultant slip, the line along the rail the longitudinal slip, and the line drawn transversely of the rail the lateral slip. It is apparent that the magnitude of the lateral component of the force producing slipping is dependent upon the relative magnitudes of the longitudinal and lateral slip. For the conditions at (a) the lateral component or lateral pressure against the rail will be small. For equal slip in the two directions as at (b), which may be the condition for a short truck like a four-wheel car truck, the lateral component will be 0.7 of the force required to overcome friction. For a longer locomotive frame (c) will represent the conditions.

Since the design of locomotives differs so greatly in detail even among those of the same type, and since there are so many uncertain and indeterminate features in the direction and relative magnitudes of the slip and in the coefficients of friction, it does not seem profitable to attempt to make an analytical treatment of the lateral forces developed at the several wheels by the turning of the locomotive, though it would be easy

to show that the magnitude of the lateral forces developed is necessarily considerable. Instead, it seems better to consider the data of the stresses observed in the rails themselves and to discuss these data in the light of the action here outlined. It may be added here that analytical considerations indicate that for the same superelevation and at low speeds the degree of curve has little effect on the nature and magnitude of the turning forces beyond a lower limit of curvature where the assumed condition comes into action and within an upper limit where other elements may exert a controlling influence.

It should be pointed out here that these turning forces, together with the centrifugal force and the component due to the transverse inclination of the track and the force required to produce lateral slip, and including the reacting force at the inner rail, are the principal and

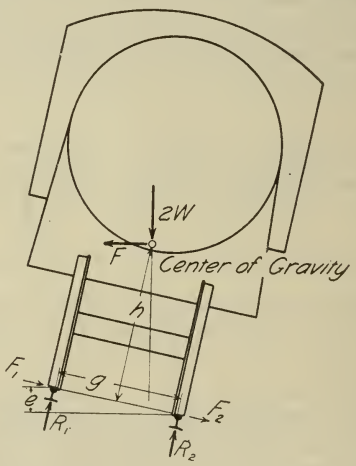


Fig. 40.—Forces and Reactions on Curve Track Having Transverse Inclination.

most important lateral forces producing bending moment and bending stress in the two rails of curved track.

(c) Division of Load by Reason of Transverse Inclination of Track and Centrifugal Force.—On straight track having the rails at the same level transversely, it may be presumed that the load on one axle is transmitted in equal amounts to the two rails. On curves an equal division between the two rails may not be generally expected. At low speeds (the centrifugal force being negligible), the superelevation of the outer rail results in more than one-half of the load being transmitted to the inner rail, with a corresponding decrease in that transmitted to the outer rail. At the higher speeds the centrifugal force acts in an important way to increase the proportion taken by the outer rail and decrease that

on the inner rail. Both of these agencies enter also into the development of lateral pressures against the two rails.

In Fig. 40, g is the distance between bearing points of the wheels on the two rails (taken in the calculation as 59 in. for the curved track used in the tests); e is the superelevation of the outer rail; and h is the distance of the center of gravity of the locomotive from the level of the top of the rails. Use W as the weight which would be applied to the rail through one wheel for straight and level track, and R' the reaction on the outer rail and R'' that on the inner rail which would be expected from analytical considerations to correspond to the weight W. Then if the speed is such that the centrifugal force is negligible, it may be shown that the load transmitted to the outer rail will be given approximately by the following equation, the cosine of the angle of the transverse inclination of the track being taken as unity,

$$R' = W\left[1 + 2\frac{h}{g}\frac{e}{g}\right] \quad \ldots\ldots (29)$$

and that transmitted to the inner rail

$$R'' = W\left[1 - 2\frac{h}{g}\frac{e}{g}\right] \quad \ldots\ldots (30)$$

For $h = 72$ in. and $e = 6$ in., these equations indicate that for low speeds the load transmitted to the outer rail is 25 per cent less than that on track level transversely and that to the inner rail is 25 per cent greater than the normal load. It is evident that the inequalities in the division of load between the two rails may be a matter of some consequence.

Equations (29) and (30) were derived on the assumption that the body of the locomotive retains the same position with respect to the axles of the drivers that it has on straight track. If as may be the case the greater weight on the inside of the curve causes the springs on that side of the locomotive to deflect more and those on the outside of the curve less than would normally be found, the result will be to tilt the locomotive still further and thus to throw greater weight on the inner rail than is given by these formulas. The effect of this tilting upon the transfer of load is less accordingly than that due to transverse inclination of track alone, since the tilting action is about the point of attachment of the equalizing levers to the frame of the locomotive, which is somewhat above the level of the axles of the drivers. Since the springs deflect less proportionately for an overload than for a small load, the effect of this change in deflection of the springs will not be as great as would otherwise be expected. It, however, may need some consideration.

For the higher speeds the centrifugal force tends to counteract this inequality. For the speed corresponding to the superelevation given to the track the inequality vanishes and it would be expected that the rails will receive equal loads. For still greater speeds the outer rail will receive the greater load.

In any case the load on a rail would be expected to be the algebraic sum of that due to equation (29) or (30) and that due to the centrifugal force. The centrifugal force at V miles an hour on a D° curve due to the weight 2W is approximately $F = \frac{2WV^2D}{85\ 800}$ Consider this force to be applied at the center of gravity of the locomotive. The reaction of the outer and inner rails, respectively, will then be

$$R_1 = W\left[\left(1 - 2\frac{e}{g}\frac{h}{g}\right) + \frac{2V^2D}{85\ 800}\left(\frac{h}{g} + \frac{1}{2}\frac{e}{g}\right)\right] \quad \ldots\ldots (31)$$

$$R_2 = W\left[\left(1 + 2\frac{e}{g}\frac{h}{g}\right) - \frac{2V^2D}{85\ 800}\left(\frac{h}{g} - \frac{1}{2}\frac{e}{g}\right)\right] \quad \ldots\ldots (32)$$

If F_1 is the resulting lateral reaction or pressure against the outer rail of the curved track and F_2 that against the inner rail, the sum of the lateral reactions for the two rails will be approximately

$$F_1 + F_2 = \frac{2WV^2D}{85\ 800} - 2W\frac{e}{g} \quad \ldots\ldots (33)$$

The first term of the second member of the equations is the centrifugal force and the second the horizontal component of the load due to superelevation of the outer rail. At the speed corresponding to the superelevation $F_1 + F_2 = 0$. It is apparent that the division of the total lateral force between F_1 and F_2 is indeterminate, as in the case of the vertical reactions of the two hinges of a door. Generally a low or a smaller resistance to lateral slip on one wheel may be expected to cause a greater part of the total lateral force to go to the other rail, except as modified by the bearing of the flange of the wheel against the side of the rail. It will be seen from equation (33) that the value of $F_1 + F_2$ for the highest superelevation of the track used in the tests, 8½ in. is 14 per cent of the total load on the axle.

Values of the ratio of the vertical load on the outer and inner rail of curved track to the corresponding value on straight track, as found by equation (31) and (32), are given in Tables 21 and 22 for the curves and speeds used in the tests.

(d) Spreading Action of Drivers and Other Wheels.—The stiffness of the frame and the smallness of the lateral play in the journals when the locomotive is just out of the shops may leave little individual freedom of lateral movement in the drivers. With a long and rigid wheel base it may be thought that on sharp curves a flange of an intermediate driver will be crowded against the inner rail while the flanges of the end drivers are pressed against the outer rail, and thus that there would be a marked tendency to spread the rails apart, this action creating lateral bending forces against the two rails. As the gage of the wheels is less than the gage of the track (the difference ranging from ¾ to 1 in. for new wheels and sometimes running up to perhaps 2 in. for badly worn flanges and

widened gage), and as on curves there may be an increase in gage over the standard, and as there may be some flexibility in the frame and bearings, it is to be expected that for a given wheel base and other stated conditions the limiting degree of curve below which no spreading action

TABLE 21

Ratio of Loads on Inner and Outer Rail—A. T. & S. F. Ry.

Locomotive	Degree of Curve	Super-elevation Inches	Speed in m.p.h.	Ratios					
				Analytical Values		Test Values			
				Inner Rail	Outer Rail	Inner Rail		Outer Rail	
Ribera									
Mountain 3710..	6	4.7	5	1.20	0.80	1.22		0.78	
			25	1.09	0.91	1.16		0.84	
			35	1.00	1.00	1.02		0.98	
			50	0.77	1.26	0.81		1.19	
Santa Fe 3813...	6	4.7	5	1.20	0.80	1.19		0.81	
			25	1.09	0.91	1.11		0.89	
			40	0.92	1.09	1.01		0.99	
Mountain 3710...	10	4.7	5	1.19	0.81	1.21		0.79	
			25	1.02	0.99	0.99		1.01	
			35	0.85	1.17	0.81		1.19	
Santa Fe 3813...	10	4.7	5	1.19	0.81	1.20		0.80	
			25	1.02	0.99	1.00		1.00	
			35	0.85	1.17	0.83		1.17	
Santa Fe 3829...	10	4.7	5	1.19	0.81	1.28		0.72	
			25	1.02	0.99	1.00		1.00	
			35	0.85	1.17	0.88		1.12	
Bealville						Up Grade	Down Grade	Up Grade	Down Grade
Santa Fe 3832...	10	4.4	10	1.16	0.84	1.15	1.15	0.85	0.85
			18	1.10	0.91	1.09	1.08	0.91	0.92
Santa Fe 3653...	10	4.4	10	1.16	0.84	1.18	1.18	0.82	0.82
			18	1.10	0.91	1.13	1.12	0.87	0.88
Cajon									
Santa Fe 3832, fourth driver cylindrical.....	10	5.4	10	1.20	0.80	1.27	1.36	0.73	0.64
			18	1.14	0.86	1.17	1.30	0.83	0.70
Santa Fe 3832 fourth driver grooved.....	10	5.4	10	1.20	0.80	1.32	1.32	0.68	0.68
			18	1.14	0.86	1.18	1.24	0.82	0.76

of the kind here meant exists will be rather large. For the purpose of giving greater freedom the flange is sometimes omitted from one or more drivers, making so-called blind drivers.*

To find whether a given curve is below the limiting value for a given wheel base on the basis that the flanges bear against the two rails

*For a discussion bearing on the widening of gage on curves, see Report of Committee V—on track, Proceedings, A. R. E. A., Vol. 23, p. 575.

TABLE 22

Ratio of Loads on Inner and Outer Rail—D. L. & W. R. R.

Locomotive	Degree of Curve	Super-elevation Inches	Speed in m. p. h.	Ratios			
				Analytical Values		Test Values	
				Inner Rail	Outer Rail	Inner Rail	Outer Rail
Mikado, 1220.....	4	3.7	5	1.16	0.84	1.03	0.97
			25	1.08	0.92	1.01	0.99
			35	1.01	0.99	0.97	1.03
			45	0.92	1.08	0.89	1.11
Pacific, 1131.....	4	3.7	5	1.16	0.84	1.11	0.89
			25	1.08	0.92	1.04	0.96
			40	0.97	1.03	.94	1.06
			60	0.74	1.28	1.00	1.00
Mikado, 1220.....	6	8.5	5	1.36	0.64	1.37	0.63
			25	1.26	0.75	1.30	0.70
			35	1.16	0.86	1.20	0.80
			45	1.02	1.01	1.16	0.84
Mikado, 1220.....	7½	6.4	5	1.27	0.73	1.29	0.71
			25	1.14	0.87	1.19	0.81
			35	1.01	1.00	1.10	0.90
			45	0.84	1.19	0.97	1.03
Pacific, 1131.....	7½	6.4	5	1.27	0.73	1.26	0.74
			25	1.14	0.87	1.16	0.84
			40	0.93	1.09	0.96	1.04
			50	0.74	1.30	0.87	1.13

in the manner just outlined, the ordinates for the curve may be found. In Fig. 41 the flanges of the outer end drivers are both considered to bear against the outer rail at points which may be somewhat beyond the limits of the wheel base, and the flanges of intermediate wheels against the inner rail, and the ordinates a and b found. Then if g is the gage of the track and g' the gage of the wheels, the distance $a - b$ must be enough greater than $g - g'$ to give reasonable clearance. For five drivers on a side, the term b vanishes if the middle driver is flanged; with a blind

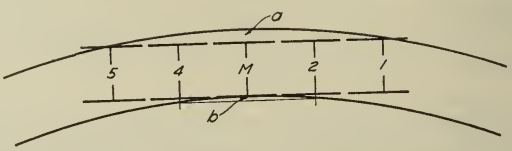


Fig. 41.—MID-ORDINATES OF PORTION OF CURVE BETWEEN CERTAIN DRIVERS OF A LOCOMOTIVE.

driver at the middle the chord from the second to the fourth driver will be used in getting the value of b.

The assumption that the outer rear driver bears against the outer rail is not warranted by the observations and tests made on curved track; the

rear driver generally keeps well away from the outer rail at low speeds and usually also at the higher speeds. This position of the rear driver in effect lengthens the wheel base and decreases the degree of curve for which clearance would be found sufficient if there were not other elements entering into the phenomenon. An important modification of the assumptions is that the lateral deflection of the rails at some of the wheels is found to be considerable, particularly in the inner rail at a rear intermediate driver. With the Mountain type locomotive on a 10° curve the inner rail was found to deflect inwardly of the curve at the third driver as much as 0.5 in., the gage of the track increasing to more than that extent because the outer rail also deflects somewhat. The Santa Fe type locomotive with all drivers flanged and the Santa Fe type with the main driver flangeless appeared to traverse the 10° curves without the flanges of the intermediate drivers bearing against the inner rail, and hence there was no spreading action of the nature of that usually considered when this subject is discussed. As a very heavy thrust against the inner rail on 10° and even on 6° curves is found, really an excessive pressure, it is plain that some other explanation of its source than the spreading action of the flanges must be sought for.

The explanation that appears plausible is connected with the change of direction of the locomotive. Consider in Fig. 38 that the force changing the direction of the locomotive is applied by the rail to the flange of the outer first driver at A, or by the wheels of the front truck at some point ahead of this. Consider that the rotation is about the point B on the inner rail and that the rear outer driver is slipped outwardly at C. The lateral pressure at B will have to resist the two forces at A and C, except as other wheels may assist. As the other wheels slip laterally, if the wheel at B does not slip, as seems to be the case, the main part of the resisting lateral thrust will be taken at B. This seems to be the more plausible, when, as is found in most tests at low speed, the load transmitted by the driver in question is greatly in excess of that taken by the others. It will be found also from some of the tests with the Santa Fe type locomotive that the conditions of this lateral thrust change materially when all ten drivers are flanged and when four of them are flangeless. Measurements of the lateral deflection of the inner and outer rails of curved track confirm the action of the lateral forces referred to above and show a considerable increase in gage without the flanges touching the inner rail.

Altogether, it seems evident that the limiting conditions of curves can not be determined by a study of ordinates and clearances alone. The absence of wear on the gage side of the inner rail of such curves of itself would seem to absolve the flanges of the drivers from spreading action on the inner rail. The presence of very great lateral pressures against the inner rail is quite evident in the tests. These pressures produce a widening of the gage, something like a spreading action, but the origin of the pressure should be attributed to other sources than lack of clearance of

flanges of the intermediate drivers. It seems clear that the lateral force is transmitted by the drivers to the inner rail by means of the friction between the tread of the driver and the head of the rail. The very high vertical pressures transmitted by one or more of the inner drivers and the greater value of static friction combine in many cases to make the point of contact of the rear intermediate driver on the inner rail the center of rotation of the change of direction of the locomotive. Thus this driver is kept from slipping laterally, while at the same time the great lateral pressure on the inner rail developed at this driver causes the lateral deflection and spreading of the inner rail to be considerable even on track of moderate curvature. In general it is evident that the method of calculating clearances indicated in Fig. 41 is not generally applicable.

(e) Other Sources of Lateral Pressure.—When the locomotive is pulling a load around a curve the tractive pull of the locomotive at the coupling with the tender will not be in the direction of the tangent to the curve at that point—in other words, there will be obliquity of traction. This obliquity of traction will produce a lateral force on the rear end of the locomotive itself, inwardly of the curve, a force that is the component of the tractive force in the direction of the radius of the curve. The approximate direction of the frame of the locomotive may be found by considering that the frame is tangent to the curve at the point of rotation of the turning movement of the locomotive around the curve that has already been referred to. The point of rotation has been found to be at or near the next to the last inner driver. For the Santa Fe type locomotive on a 10° curve the angle of the direction of the locomotive with the tangent to the curve would then be not more than 2°, and probably less, and for the other locomotives and for lighter curves the angle would be smaller still. It will be seen that under these conditions the lateral component of the tractive force may be expected to be small in comparison with the other loads and forces acting. The effect of this lateral component applied at the point of the coupling will be to tend to swing the rear end of the locomotive inwardly of the curve and the front end outwardly. The effect at the rear drivers will be to relieve the outward lateral thrust on the rail at that point if these drivers do not slip outwardly, and to increase the lateral force acting on the locomotive at that point if the rear drivers do slip outwardly. Both conditions would tend to increase the lateral inward thrust of the driver at the point of rotation and also the outward lateral thrust at the front driver or front truck wheel. From the small value of the angle of obliquity and the possible magnitude of the tractive force it seems very probable that the lateral thrusts due to obliquity of traction are all small. Whether they actually have any serious bearing on the lateral bending of the rail may best be found from the experiments. It may be added that Wellington in the discussion of the effect of obliquity of traction on train resistance came to the conclusion that the additional force required to pull the train due to this cause was too small to need consideration.

On straight track there are ordinarily lateral movements of locomotives and cars or so-called vibrations which cause lateral bending stresses in the rail. These may be due to variations in the track, or to lateral motion of the locomotives and cars. The lateral bending stresses produced in this way may be of considerable magnitude, as has already been shown. On curved track the lateral outward and inward movements may be expected to be less than on straight track because of the action of the lateral component of the load due to the transverse inclination of the track and of the lateral forces developed in the turning of the locomotive. Even at the speed of superelevation the lateral forces will be quite effective in keeping the wheels in a given position. It is true, of course, that if the curved track is not in well-kept condition of line and superelevation, the lateral movement may be much greater than on straight track, as when the locomotive and cars careen from side to side.

It should be understood that the discussion of the action of curved track here made is preliminary to the discussion of the results of the tests; no analytical treatment can have finality.

13. **General Results of Tests on Curved Track.**—In so complicated a problem, it is not easy to present the data of the tests, interpret the meaning of the results, and distinguish between the effects of the various factors which together influence the action of the locomotive and track. It is to be expected then that there may be some repetition or duplication of statement in the discussion. Space will permit the presentation of only a part of the data, but representative parts have been selected and general average values are given in the tables and diagrams.

Fig. 42 to 63 give the stresses at the inside edge and the outside edge of the base of the inner and the outer rail under the wheels of six locomotives at several speeds throughout the revolution of the driver in the tests on the A. T. & S. F. Ry. and the D. L. & W. R. R. The observations from which these averages were obtained when plotted with respect to the revolution of the driver were found to be somewhat more scattered than is the case with straight track, as is to be expected from the greater number of variable factors entering into the problem, but the body of the observed values lie in a belt that is frequently within 6,000 lb. per sq. in. of stress of the line of average stress, though in some cases the range is much wider. This is true for diagrams representing mean stress, stress at inner edge, and stress at outer edge of base of rail. It is evident then that as in the case of straight track, the line or curve of average stress may be taken to be representative of the values, with the understanding that values may frequently be expected greater and less than this line of average stress within the limits of the belt, with occasional values greater or less than these limits, and that the range of stress above and below the average line may be taken to be 4,000 to 12,000 lb. per sq. in. for the rail sections and loads used in the tests.

In Fig. 64 to 67 are given the stresses at inside edge and outside edge of the base of the inner and the outer rail under the several wheels of three types of locomotives at the speeds used in the tests.

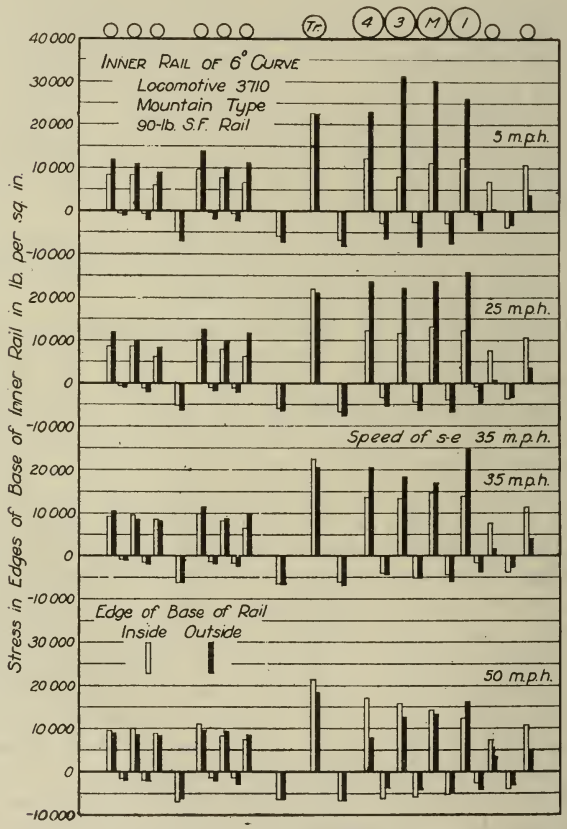


Fig. 42.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 6° Curve—Series 5319-37, Mountain Type Locomotive of the A. T. & S. F. Ry.

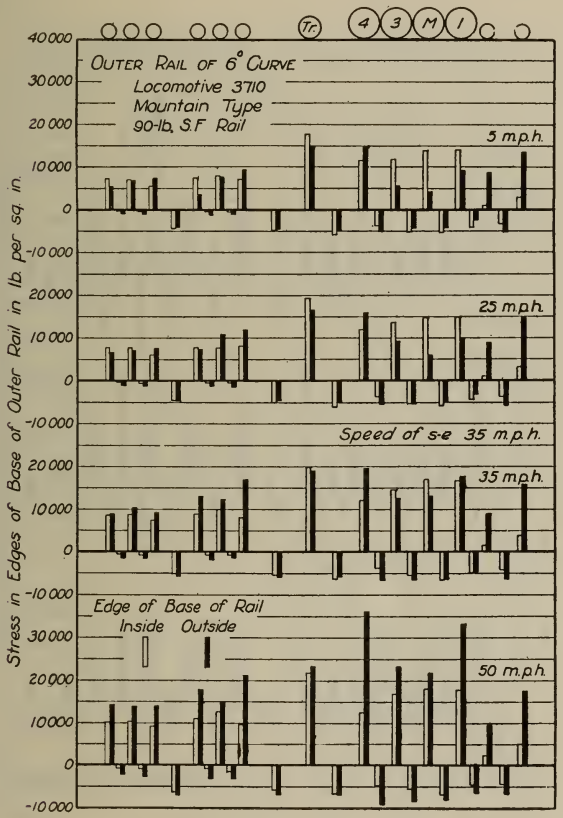


Fig. 43.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 6° Curve—Series 5319-37, Mountain Type Locomotive of the A. T. & S. F. Ry.

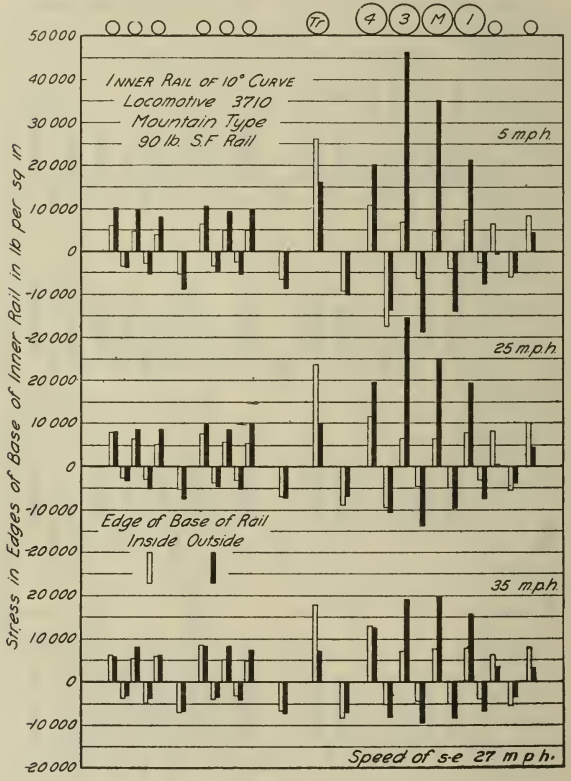


Fig. 44.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 10° Curve—Series 5351-65, Mountain Type Locomotive of the A. T. & S. F. Ry.

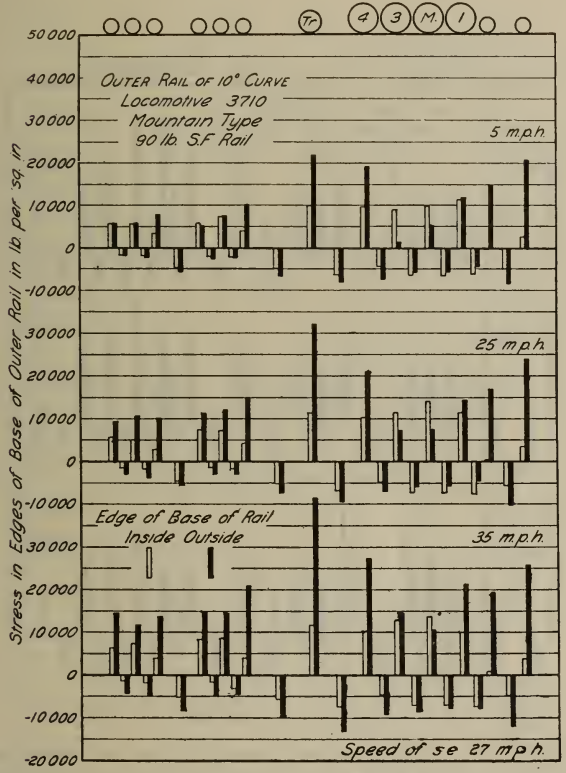


Fig. 45.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 10° Curve—Series 5351-65, Mountain Type Locomotive of the A. T. & S. F. Ry.

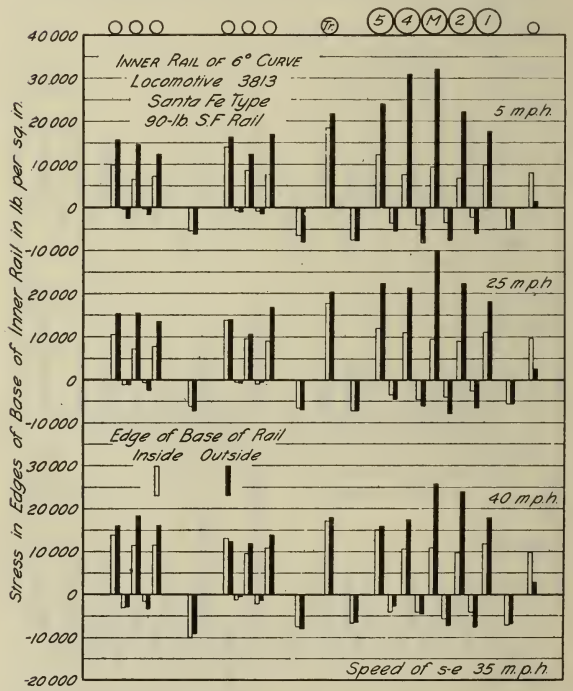


Fig. 46.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 6° Curve—Series 5383-5400, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

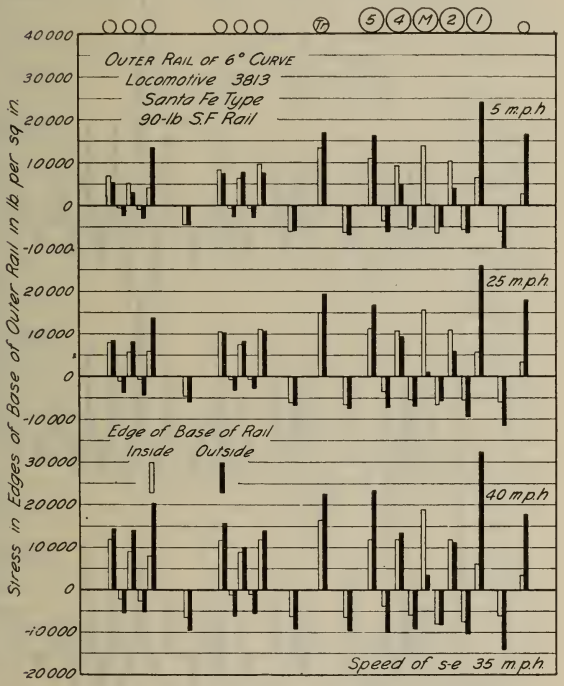


Fig. 47.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 6° Curve—Series 5383-5400, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

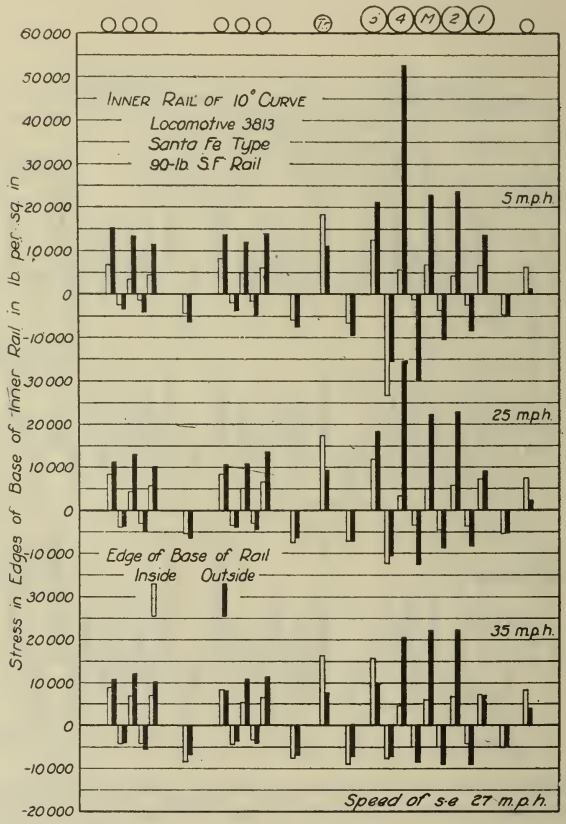


Fig. 48.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 10° Curve—Series 5366-82, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

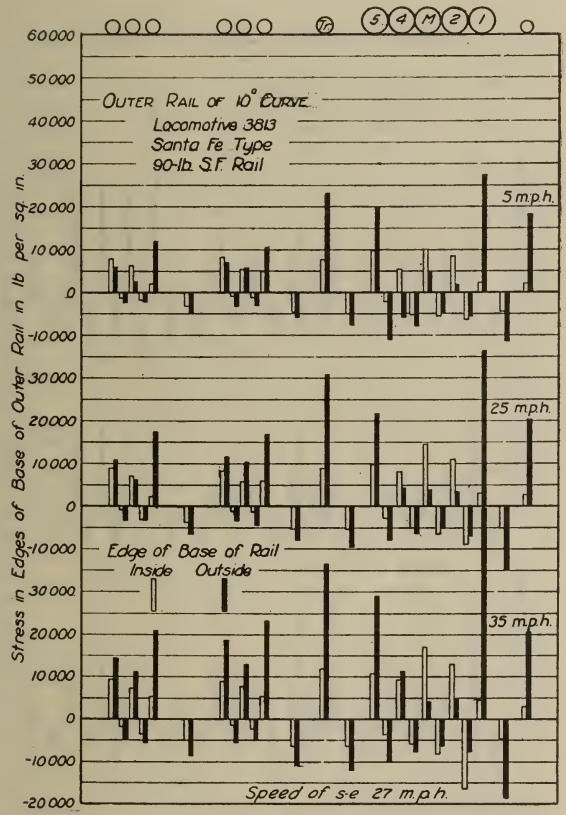


Fig. 49.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 10° Curve—Series 5366-82, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

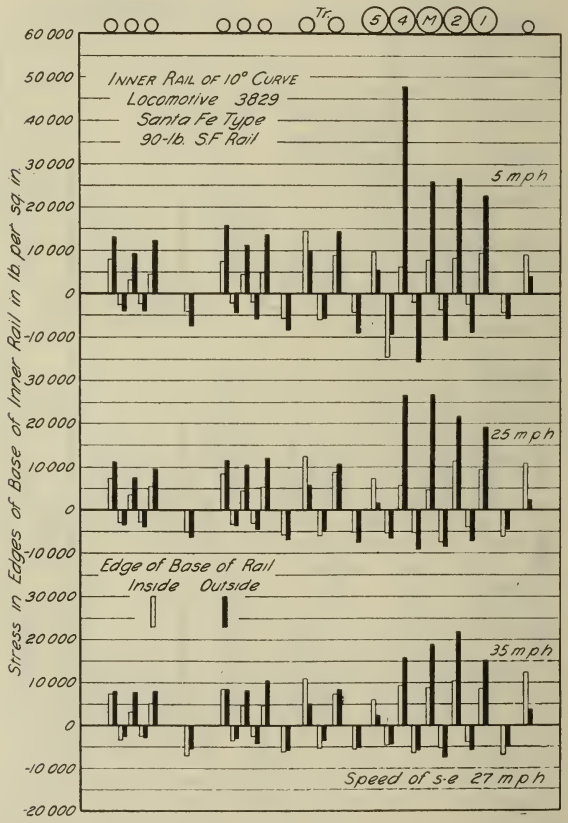


Fig. 50.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 10° Curve—Series 5425-33, Double Trailer Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

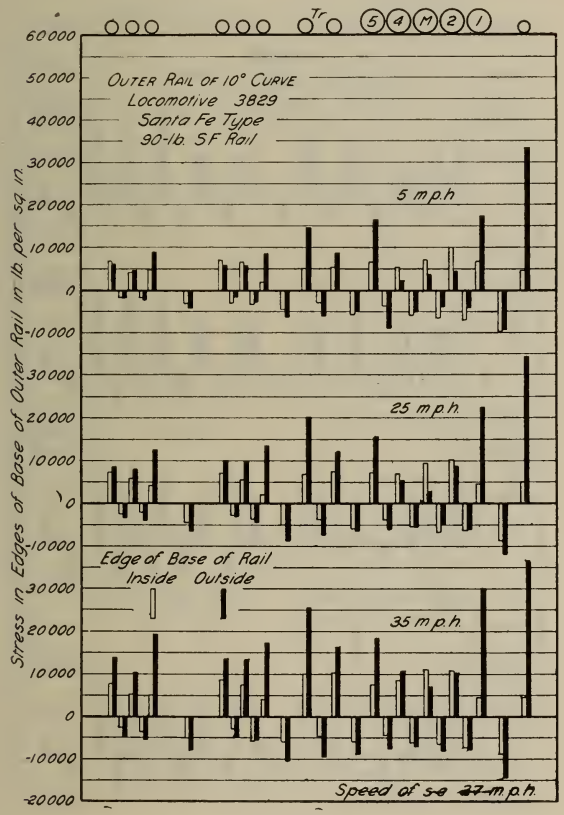


Fig. 51.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 10° Curve—Series 5425-33, Double Trailer Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

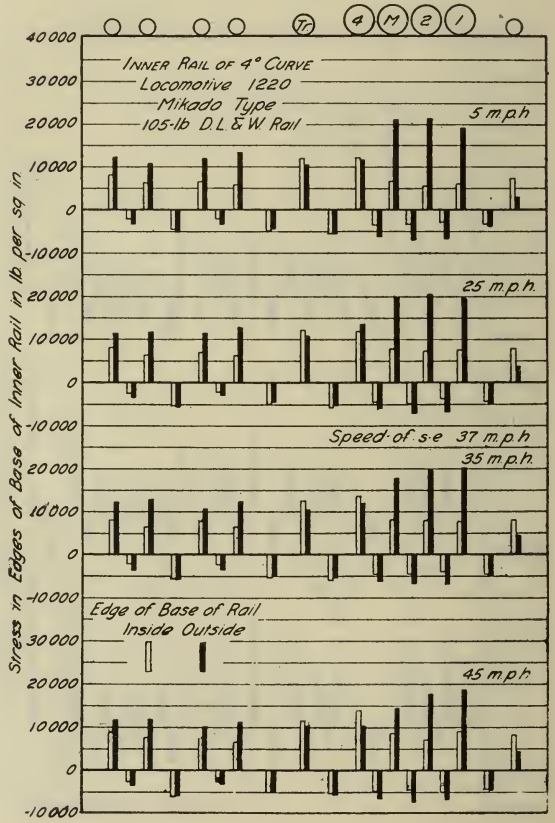


Fig. 52.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 4° Curve—Series 5150-69, Mikado Type Locomotive of the D. L. & W. R. R.

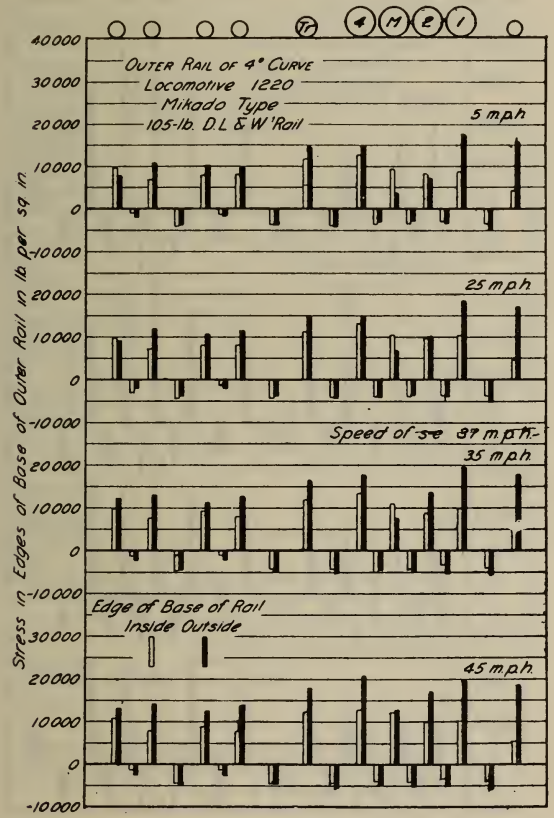


Fig. 53.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 4° Curve—Series 5150-69, Mikado Type Locomotive of the D. L. & W. R. R.

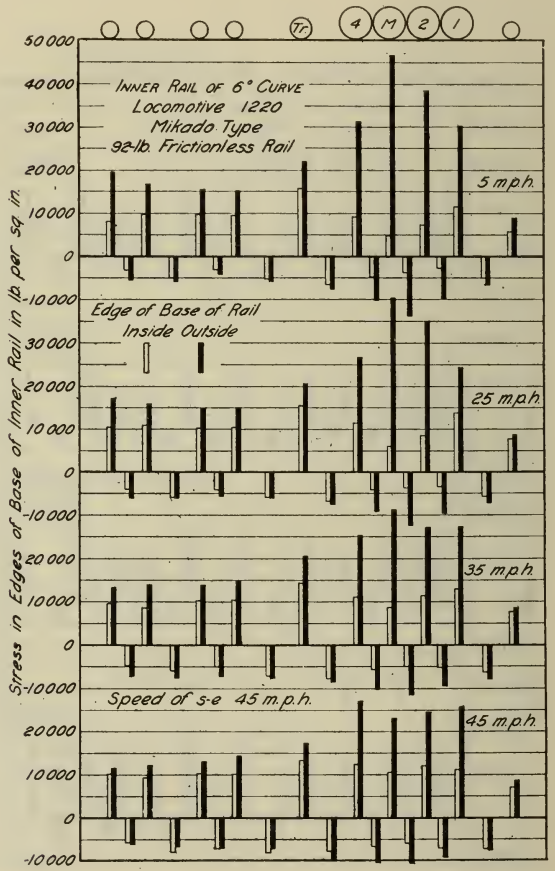


Fig. 54.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 6° Curve—Series 5191-5205, Mikado Type Locomotive of the D. L. & W. R. R.

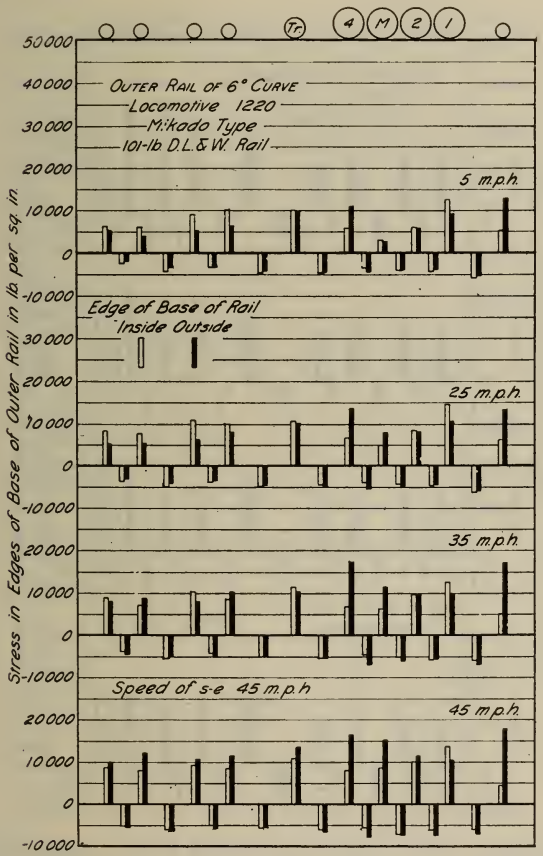


Fig. 55.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 6° Curve—Series 5191-5205, Mikado Type Locomotive of the D. L. & W. R. R.

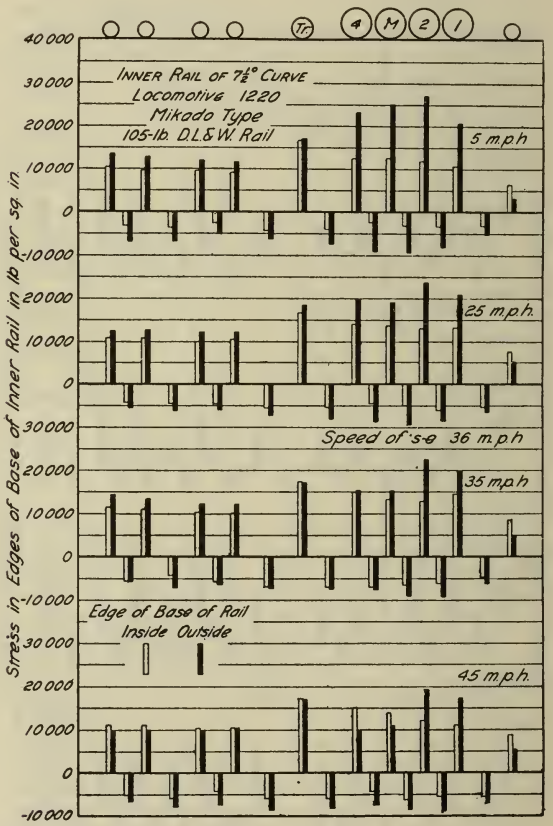


Fig. 56.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 7½° Curve—Series 5170-90, Mikado Type Locomotive of the D. L. & W. R. R.

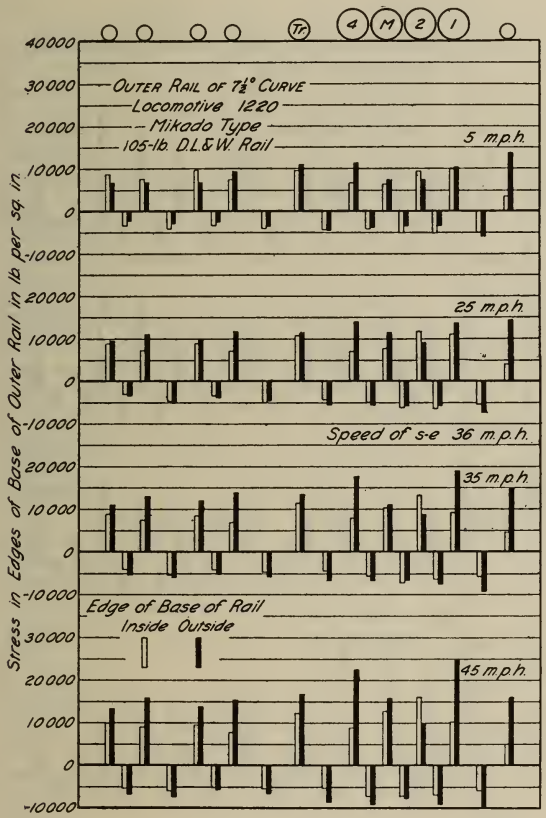


Fig. 57.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 7½° Curve—Series 5170-90, Mikado Type Locomotive of the D. L. & W. R. R.

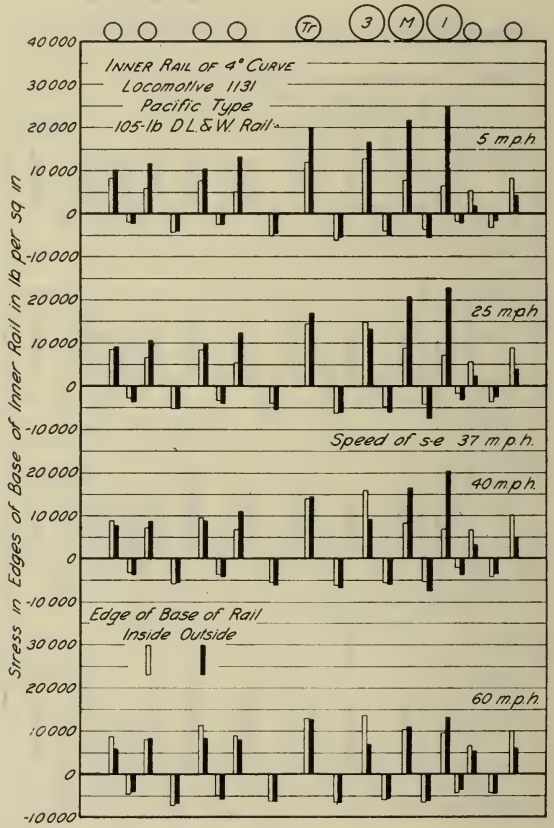


Fig. 58.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 4° Curve—Series 5100-16, Pacific Type Locomotive of the D. L. & W. R. R.

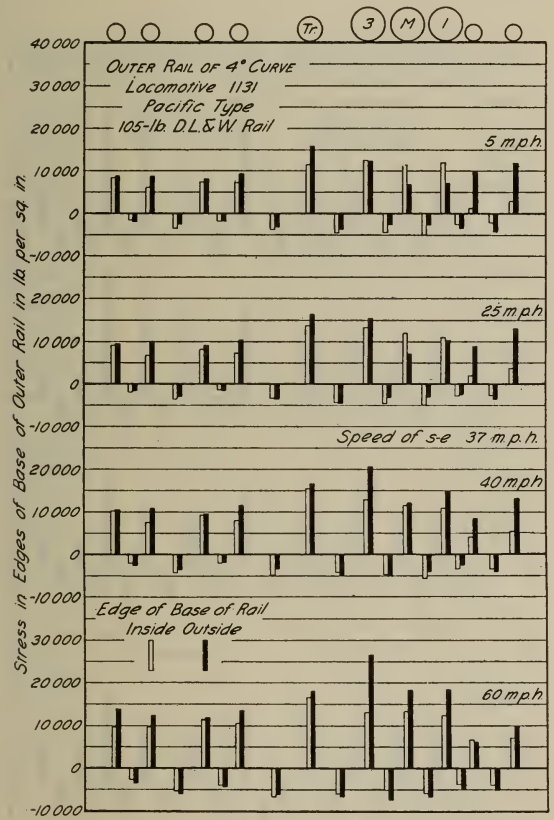


Fig. 59.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 4° Curve—Series 5100-16, Pacific Type Locomotive of the D. L. & W. R. R.

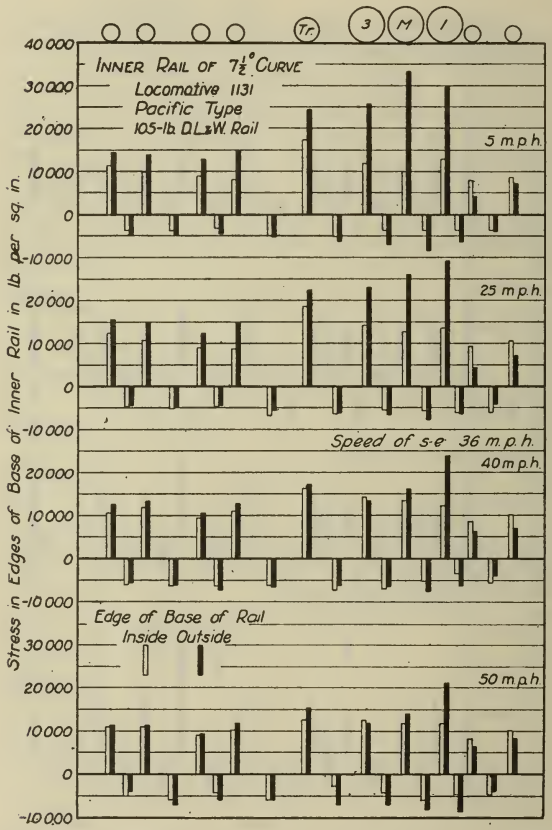


Fig. 60.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 7½° Curve—Series 5117-35, Pacific Type Locomotive of the D. L. & W. R. R.

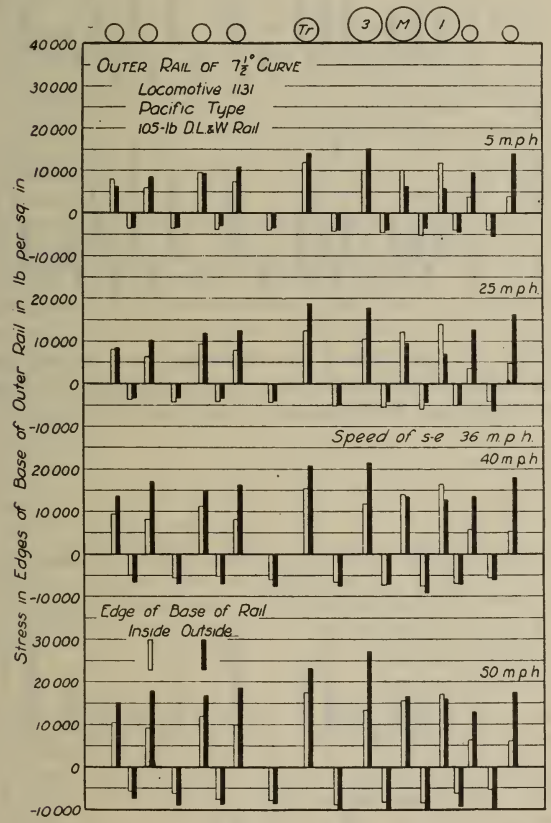


Fig. 61.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 7½° Curve—Series 5117-35, Pacific Type Locomotive of the D. L. & W. R. R.

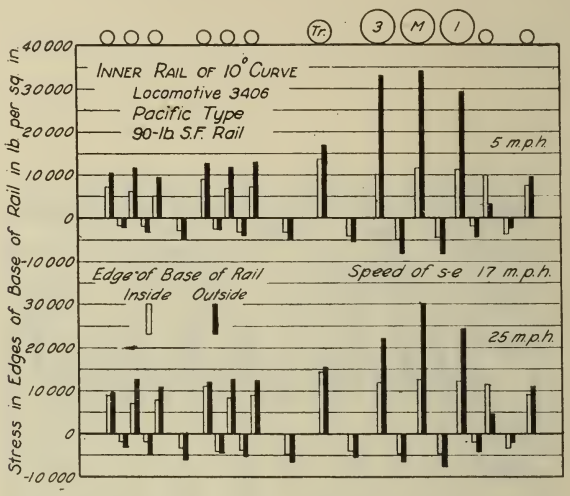


Fig. 62.—Stress at the Inside and Outside Edges of the Base of the Inner Rail of the 10° Curve at Ft. Madison, Ia.—Series 5459-66, Pacific Type Locomotive of the A. T. & S. F. Ry.

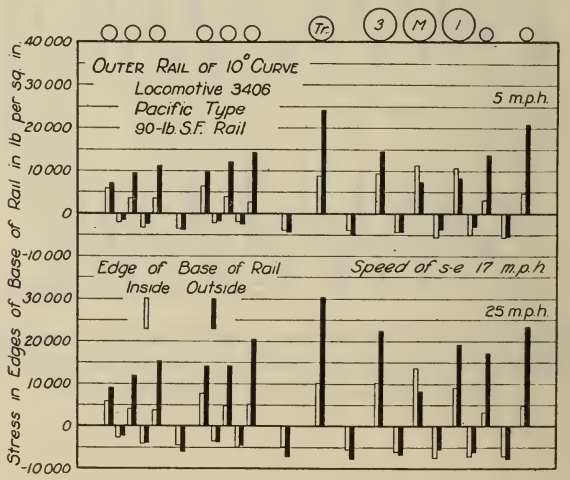


Fig. 63.—Stress at the Inside and Outside Edges of the Base of the Outer Rail of the 10° Curve at Ft. Madison, Ia.—Series 5459-66, Pacific Type Locomotive of the A. T. & S. F. Ry.

In making comparisons between tests reference may be made to the properties of the rail sections given in Table 1 and the weights of the locomotives given in Fig. 1 to 3.

From an examination of the diagrams it is evident that there are marked differences in stresses under the several drivers, that these differences vary greatly with speed, and that there are high lateral bending stresses in the rail under certain wheels.

14. **Stresses Due to Vertical Bending of the Rail.**—The mean stress in the base of rail (the average of the stresses observed at the two edges and generally hereafter called the vertical bending stress) may be taken as representative of the bending of the rail in a vertical plane, or more strictly of the bending in a direction normal to the inclination of the track, the two effects being practically identical for the superelevation of the track used in the tests since the cosine of the angle of inclination is very close to unity. Although stresses in rail are not exactly proportional to the loads when the same total load is differently distributed among the several wheels or differently divided between the two rails, yet when the differences are not great the sum of the stresses in the rail under all the wheels will generally not differ greatly for different divisions or distribution of load, and summations of the stresses for each rail may be useful in making comparisons and in checking up on the action of the locomotive and track. By means of the analytical method given in the first progress report and the solution for unknowns in equations based thereon, the load which will produce the given mean stress may be estimated quite approximately. It will be seen then that the values of the mean stresses in base of rail (vertical bending stresses) not only will permit comparisons with the stresses in straight track to be made, but will enable the vertical load to be distinguished from the lateral forces or loads that are also developed in passing around the curve. A close estimate may then be made of the distribution of vertical load among the wheels and between the two rails as it is found on the various curves. A discussion will be given first based on the average stresses in the two rails and then one based on the loads at the individual wheels and the stresses in the rails.

Fig. 68 gives the average of the vertical bending stresses in the inner and the outer rails for the Mountain type locomotive on the 6° curve of the A. T. & S. F. Ry. and also the average vertical bending stress in base of rail found on straight track with the same locomotive. These stresses are the average of the stresses found throughout the revolution of the driver. It is seen that the average of the vertical bending stresses for the inner and outer rail does not differ greatly from the stress found on straight track. Generally speaking, in most of the tests on the curved track the sum of the vertical bending stresses under all of the wheels was nearly that found on straight track, though in some cases it was somewhat larger on the curved track. It is evident that the total of the vertical

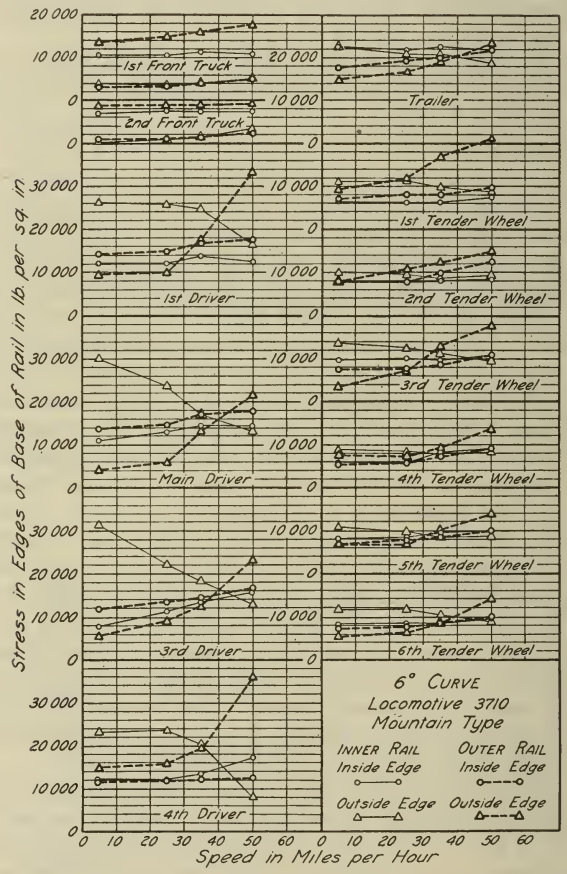


Fig. 64.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 6° Curve with Mountain Type Locomotive of the A. T. & S. F. Ry., at the Several Speeds.

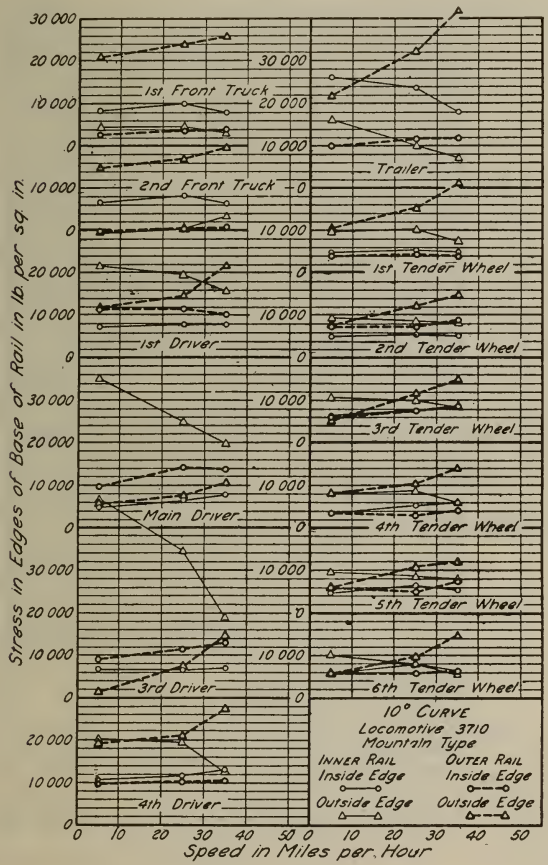


Fig. 65.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve with Mountain Type Locomotive of the A. T. & S. F. Ry., at the Several Speeds.

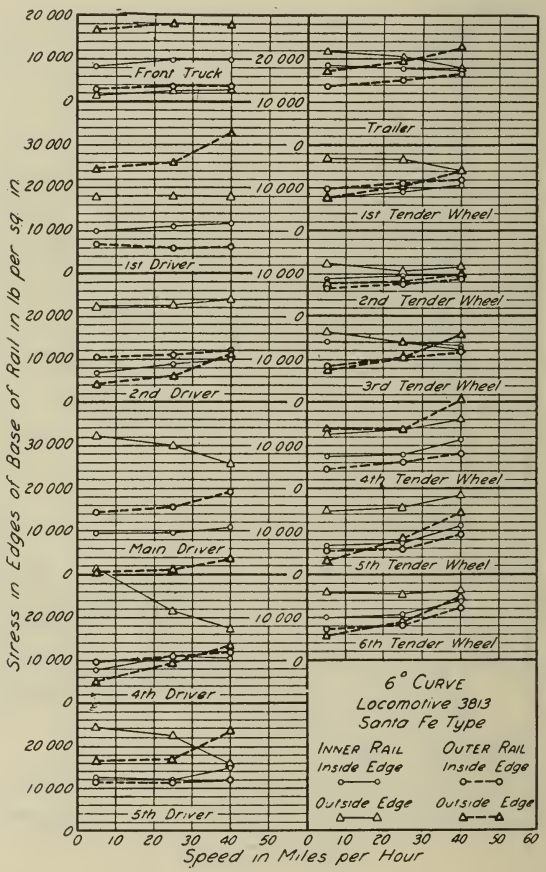


Fig. 66.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 6° Curve with Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry. at the Several Speeds.

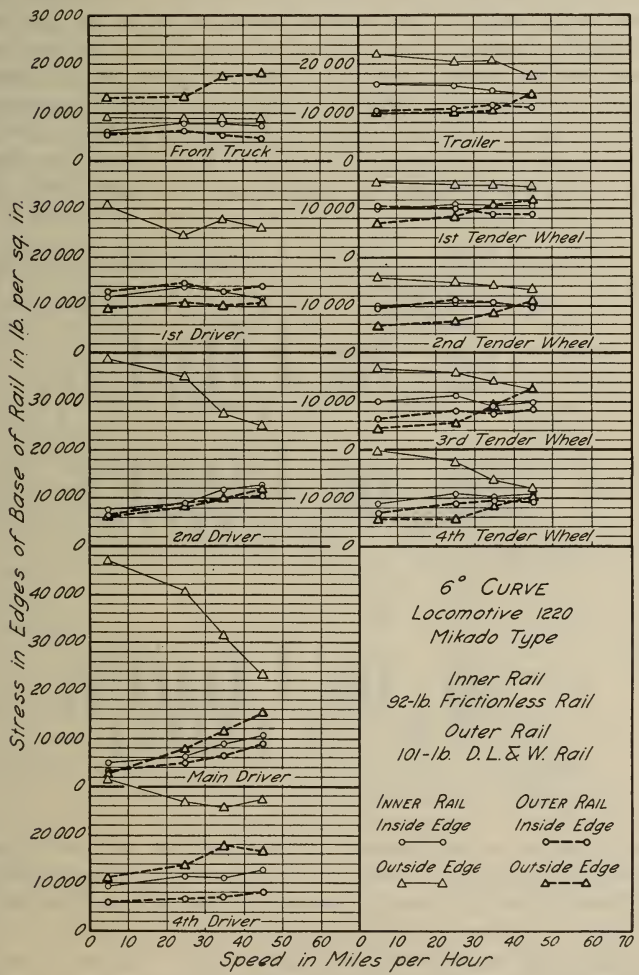


Fig. 67.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 6° Curve with Mikado Type Locomotive of the D. L. & W. R. R. at the Several Speeds.

load effects does not differ greatly in the two kinds of tracks.

It may be interesting to find whether the load is divided between the two rails in the manner indicated by equations (31) and (32), given in Article 12, The Action of Curved Track. For this purpose the sum of the vertical bending stresses in base of rail for each of the drivers, trailer and front truck wheel on one rail was taken, and also the sum of the vertical bending stresses in the base of the other rail. The ratio found by dividing the sum in the outer rail by the average of the two sums for the various tests is given in Tables 21 and 22, as is the corresponding ratio for the inner rail. In a later discussion it is shown that the sum of the stresses is closely representative of the relative loads on the two rails. If the sum of the stresses found on straight track was used as the divisor,

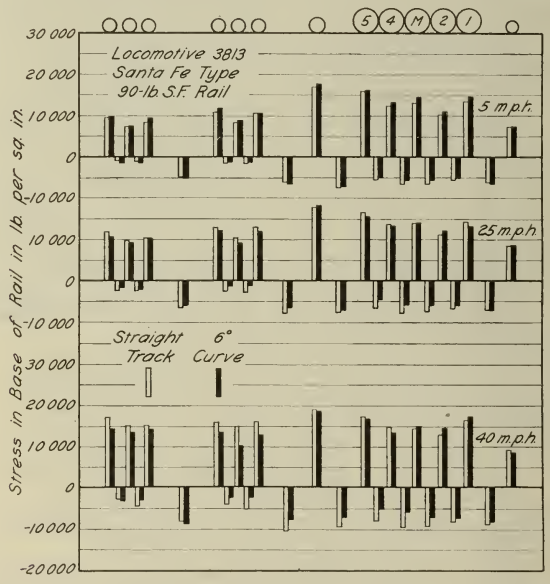


Fig. 68.—Values of the Average of the Stresses in the Inner and Outer Rail of the 6° Curve and the Stress on Straight Track —Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

the resulting ratios would have quite similar values. The ratios are given in the tables for the various speeds at which tests were made. The ratios calculated by equations (31) and (32) are also given in the tables.

For the lower speeds it is seen that the division of stress between the two rails (and presumably also the division of load) for the Pacific type on the 4° and 7½° curves of the D. L. & W. R. R. and the Mountain type and Heavy Santa Fe type on the 6° and 10° curves of the A. T. & S. F. Ry. agree very closely with the ratios calculated

from the inclination of track and centrifugal force, thus confirming the analysis. The stresses observed under the Mikado type on the 4° curve at the lower speeds give ratios which are much closer to unity than the calculated ratios; it is not known why these differ; for the 7½° curve the agreement is close. The Pacific type of the A. T. & S. F. Ry. on the 10° curve with a superelevation of 2 inches gives ratios of observed stresses which vary from unity by an amount that is about twice as great as do the calculated ratios. It appears from this that for slight superelevation curvature alone gives an effect in changing the division of load between the two rails greater than that found from the inclination of track and the centrifugal force. Evidently there is some transfer of load by other means, possibly the action of the equalizing levers.

For the higher speeds, above those corresponding to the speeds of superelevation, more variation in the results are found. The observed stresses give ratios which correspond to those obtained by analysis in the case of the Mikado type locomotive on the 4° curve, the Pacific type on the 4° curve, the Mountain type on the 6° and 10° curves, and the Heavy Santa Fe type on the 10° curves. Others give seemingly inconsistent ratios at the higher speeds; the Mikado type locomotive on the 7½° curve gives ratios derived from the test values which are larger than those obtained from the analytical values.

Since the summation of the stresses under the wheels of the locomotive for each rail may be taken to represent closely the relative loads on the inner and outer rail, it is evident that at the lower speeds, except for the 10° curve with only 2 in. of superelevation, the division of the total load between the inner and outer rails is closely represented by the calculated values based upon the transverse inclination of track and the centrifugal force. The distribution of load among the several wheels and even its division between the two wheels of one axle are quite different matters, and important variations from the distribution found on straight track will be noted.

Consideration will now be given to the vertical bending stresses under the individual wheels and the loads necessary to produce them.

The vertical bending stress in base of rail under the various wheels of the locomotive (the average stress throughout the revolution of the driver) is recorded in Fig. 69 to 72 for both outer and inner rails for the speeds used in the tests on curved track. Comparison may be made with the stresses under the same wheels in the tests on straight track, as given in Fig. 10 to 20 in Tables 10 to 18. In making a study of the stresses it should be kept in mind that changes in values of the vertical bending stresses at any wheel must be due to changes in the amount of load on the wheel either by itself or in combination with adjoining wheels.

Values of the wheel loads necessary to produce the observed vertical bending stresses may be calculated by the use of the analytical method for determining the stress in rail caused by a group of wheel loads given in the first progress report. An algebraic equation is written out for the

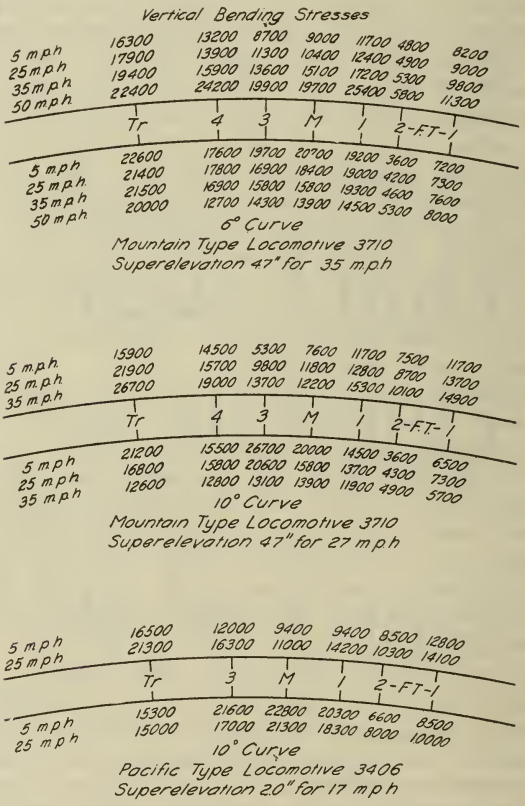


Fig. 69.—Vertical Bending Stresses in Base of Rails of Curved Track —Mountain and Pacific Type Locomotives of the A. T. & S. F. Ry.

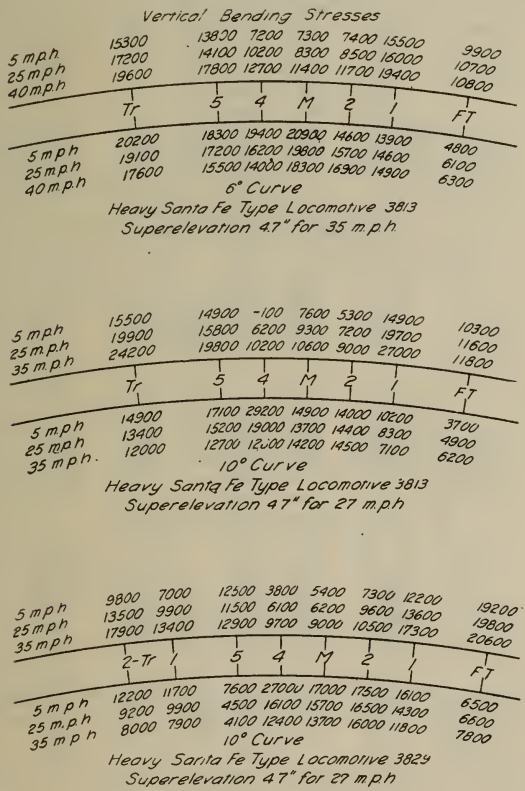


Fig. 70.—Vertical Bending Stresses in Base of Rails of Curved Track —Heavy Santa Fe Type Locomotives of the A. T. & S. F. Ry.

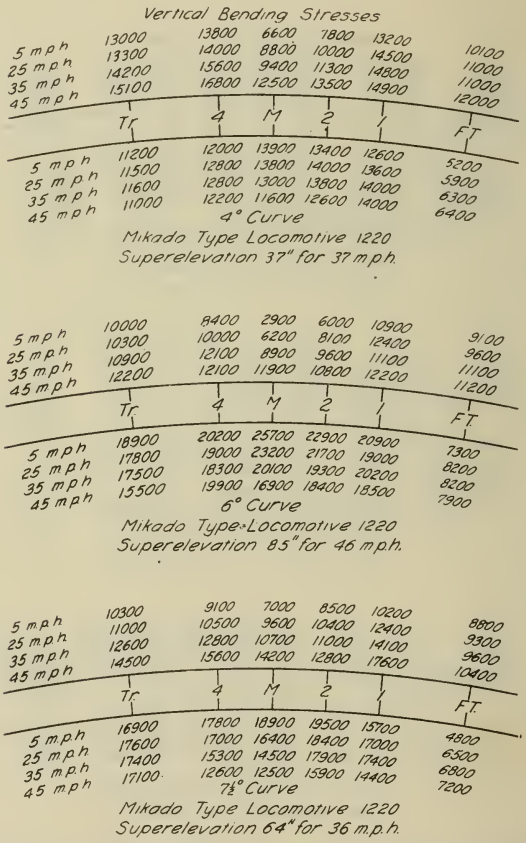


Fig. 71.—Vertical Bending Stresses in Base of Rails of Curved Track —Mikado Type Locomotive of the D. L. & W. R. R.

vertical bending stress at each driver; the observed vertical bending stress at the driver is a known quantity and the unknown wheel loads of the several drivers are represented as unknown quantities. There will then be as many equations and unknown quantities as there are drivers. The effect of the trailer and the wheels of the front truck is taken into account whenever necessary. The solution of these equations is not troublesome. The fact that the sum of all the loads thus found agrees quite closely with the total load on the drivers should give some confidence in this method of calculating the division of load among the wheels.

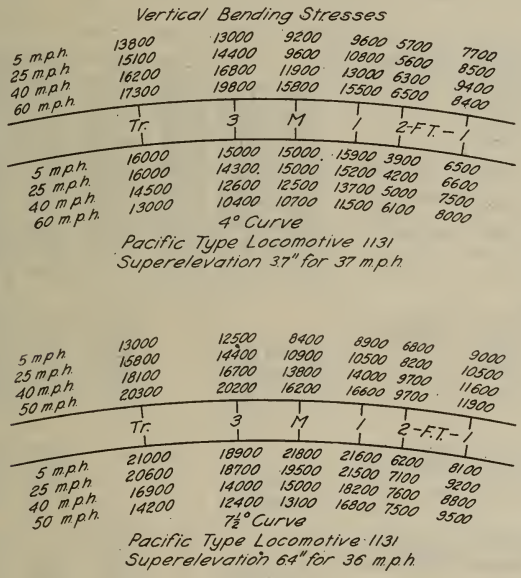


Fig. 72.—Vertical Bending Stresses in Base of Rails of Curved Track —Pacific Type Locomotive of the D. L. & W. R. R.

In Table 23 are given the calculated vertical wheel loads on straight track and on the inner and the outer rail of curves for several types of locomotive at a speed of 5 or 10 miles per hour, as determined by this method. The results of the calculations were changed proportionally to make the sum of the wheel loads equal the total load on the drivers. The values given in the table, while subject to some errer, may be taken as representative of the vertical loads transmitted to the rail. The same method may, of course, be applied to any speed.

It is evident from the diagrams and tables that the changes in vertical bending stresses and in vertical loads at the several drivers are very marked and that important additions to the bending moments in the rail

and in the bearing pressures at the top of the rail are found on curves. The increased stress in the inner rail under certain drivers is greatest at the low speed, but even at speeds two-thirds the speed corresponding

TABLE 23

Vertical Loads on Drivers Calculated from Observed Vertical Bending Stresses for Speed of Five and Ten Miles per Hour. The nominal load on the drivers is also given.

Type of Locomotive, Degree of Curve and Location	Driver Number				
	5	4	3	2	1
Heavy Santa Fe Type:					
Nominal loads on straight track	32,000	31,200	31,500	29,800	30,200
Calculated loads on straight track	32,000	33,000	33,000	29,000	28,000
6° Curve, Ribera — Outer Rail	25,000	20,000	19,000	21,000	29,000
6° Curve, Ribera — Inner Rail	37,000	46,000	47,000	37,000	29,000
10° Curve, Ribera — Outer Rail	26,000	8,000	18,000	18,000	28,000
10° Curve, Ribera — Inner Rail	41,000	64,000	45,000	37,000	24,000
10° Curve, Bealville—up-grade — Outer Rail	25,000	7,000	21,000	18,000	23,000
10° Curve, Bealville—up-grade — Inner Rail	35,000	60,000	50,000	37,000	26,000
10° Curve, Bealville—wet rails — Outer Rail	25,000	16,000	22,000	20,000	25,000
10° Curve, Bealville—wet rails — Inner Rail	32,000	45,000	44,000	41,000	32,000
10° Curve, Cajon—up-grade — Outer Rail	9,000	21,000	22,000	17,000	24,000
10° Curve, Cajon—up-grade — Inner Rail	52,000	33,000	47,000	46,000	31,000
10° Curve, S. P. Locomotive—up-grade — Outer Rail	25,000	16,000	20,000	20,000	27,000
10° Curve, S. P. Locomotive—up-grade — Inner Rail	29,000	50,000	47,000	36,000	27,000
Mountain Type:					
Nominal loads on straight track		30,200	30,200	30,400	30,700
Calculated loads on straight track		29,000	31,000	33,000	29,000
6° Curve, Ribera — Outer Rail		22,000	21,000	21,000	22,000
6° Curve, Ribera — Inner Rail		33,000	42,000	44,000	38,000
10° Curve, Ribera — Outer Rail		23,000	16,000	19,000	23,000
10° Curve, Ribera — Inner Rail		32,000	53,000	45,000	32,000
Pacific Type:					
Nominal loads on straight track		.....	30,700	29,200	29,900
Calculated loads on straight track		.....	28,000	32,000	30,000
10° Curve, Ft. Madison — Outer Rail		.....	20,000	20,000	19,000
10° Curve, Ft. Madison — Inner Rail		.....	37,000	45,000	39,000
Mikado Type:					
Nominal loads on straight track		29,600	29,600	29,600	29,600
Calculated loads on straight track		29,000	31,000	31,000	27,000
4° Curve, Dover — Outer Rail		29,000	22,000	24,000	28,000
4° Curve, Dover — Inner Rail		29,000	38,000	37,000	30,000
6° Curve, Mt. Tabor — Outer Rail		14,000	10,000	15,000	19,000
6° Curve, Mt. Tabor — Inner Rail		38,000	53,000	50,000	38,000
7½° Curve, Paterson — Outer Rail		18,000	18,000	21,000	20,000
7½° Curve, Paterson — Inner Rail		36,000	45,000	46,000	33,000

to superelevation increases of stress of 75 to 85 per cent of the increase found in the low speed were common. At the higher speeds the special increase in load at the one driver vanishes, and the distribution among the several drivers is much the same as would be expected from a consideration of transverse inclination of track and centrifugal force, except that the vertical bending stress in the outer rail under the outer front driver is very great at the high speeds.

Consider the results for the Heavy Santa Fe type locomotive given in Fig. 70. At 5 miles per hour the vertical bending stresses in the inner rail of the 6° curve under the third and fourth drivers are much

higher than the corresponding ones in the outer rail—far higher than are to be expected from considerations of transverse inclination of track and centrifugal force. In Table 23 it is seen that the loads transmitted by these inner drivers estimated from the observed vertical bending stresses are 47,000 and 46,000 lb., 50 per cent greater than the nominal loads. On the 10° curve at Ribera, at 5 miles per hour, the vertical bending stress in the inner rail under the fourth driver is 29,000 lb. per sq. in. By calculation from the stresses, the load transmitted by this driver is estimated to be 64,000 lb. (see Table 23), more than twice the normal load. The stresses in the outer rail under wheels on the same axles are correspondingly light, that on the 10° curve under the fourth driver being nothing. Considering the stresses under adjoining wheels, the load on the driver giving a rail stress of zero is calculated to be 8,000 lb., about one-fourth that acting on straight track. For both curves the high stresses in the inner rail become smaller at the higher speeds, and at 35 and 40 miles per hour (above the speed of super-elevation) they are fairly uniform under the several drivers. On the 10° curve the vertical bending stresses in the outer rail under all the drivers increase with the increase in speed. The stress of 27,000 lb. per sq. in. under the first driver at a speed of 35 miles per hour is the highest vertical bending stress found in the outer rail, and the corresponding calculated load is 47,000 lb.

In the Mountain type locomotive (see Fig. 69) the vertical bending stresses in the inner rail of the 6-degree curve under the second and third drivers at a speed of 5 miles per hour are 2⅓ and 2¼ times those in the outer rail. This means loads of 44,000 and 42,000 lb., 40 per cent more than the normal load on straight track—far more than will be produced by transverse inclination of track alone. On the 10° curve the corresponding vertical bending stress in inner rail under the second driver is nearly three times as great as that in the outer rail and under the third driver five times as great. This means loads of 45,000 and 53,000 lb., 50 per cent more than on straight track for the second driver and 75 per cent for the third driver, the latter driver thus giving a very high bearing pressure on the rail. At the speed of 35 miles per hour the stresses in the two rails of the 6° curve under the various wheels have approached each other and are nearly equal. A somewhat similar condition exists in the 10° curve for the two rails though the tendency to high stresses in the outer rail under the first and last driver is present with this locomotive as with the Heavy Santa Fe type. At the speed of 25 miles per hour, the calculated load on the third driver on the inner rail of the 10° curve is 42,000 lb. On the 6° curve at 50 miles per hour the vertical bending stress in the outer rail at the first driver is 25,400 lb. per sq. in. and that at the fourth 24,200 lb. per sq. in. The trailer of the Mountain type gives high stresses in one rail or the other, the highest value being reached in the outer rail of the 10° curve at 35 miles per hour. It is seen that except for the trailer the speed

that is higher than that corresponding to the superelevation gives the most uniform values of the stresses under all the wheels and in the two rails.

The Pacific type locomotive on the 10° curve at Fort Madison, Iowa (Fig. 69), develops a vertical bending stress in the inner rail under the first driver at a speed of 5 miles per hour that is 2.2 times that in the outer rail, and one under the second driver that is 2.4 times the corresponding stress in the outer rail. The loads necessary to produce these stresses in the inner rail are 39,000 and 45,000 lb., 1.3 and 1.5 times the normal loads on these two drivers (the driver loads that apply to straight track), although the calculated ratio of load on inner rail to normal load for the superelevation of 2 in. is only 1.08. At 25 miles per hour (the speed corresponding to the superelevation is 17 miles per hour), the vertical bending stress in the inner rail under the main driver is still nearly double that in the outer rail and the corresponding calculated loads are 40,000 and 24,000 lb. The distribution of load among the drivers at both speeds is evidently considerably different from that which obtains on straight track. It seems that for this sharp curve and small superelevation the equalizing system does not act to give an equal distribution of load among the drivers. The Pacific type locomotive on the 4° curve of the D. L. & W. R. R. at 5 miles per hour develops vertical bending stresses in the inner rail under the first and second drivers that are 1.6 times the corresponding stresses in the outer rail. These stresses indicate that the loads on these two drivers, 39,000 and 41,000 lb., are 20 and 25 per cent greater than the normal load, although the calculated ratio (and also the ratio of observed values for the locomotive as a whole) is only 1.5. On the 7½° curve at 5 miles per hour the same drivers develop vertical bending stresses in the inner rail 2.4 and 2.6 times those in the outer rail; the corresponding loads on the inner rail are 46,000 and 49,000 lb., 40 to 50 per cent greater than the normal loads, while the ratio calculated from the transverse inclination of the track is only 1.27. On both the 4° and the 7½° curve at speeds at and slightly above those corresponding to superelevation the vertical bending stresses in both rails under all the drivers of this locomotive approach common values, showing that the distribution of load is much the same as is given by the equalizing levers on straight track.

The Mikado type locomotive of the D. L. & W. R. R., which has the first and second drivers connected together by equalizing levers and also the third and fourth drivers, shows a different distribution and grouping of the stresses. The ratios of the vertical bending stress in the inner rail of the 4° curve at 5 miles per hour to stress on straight track derived by the analysis and calculation are, in the order of the drivers, 1.04, 1.22, 1.26 and 1.04. For each two drivers that are connected by equalizers, the ratio is higher for the intermediate driver than for the front or rear one, but the average is close to 1.15, the value calculated from the inclination of track. On the 7½° curve at 5 miles per hour the same transfer of load from first to second driver and from

fourth to third occurs. The ratios of stress in inner rail to stress in outer rail are 1.5, 2.3, 2.7 and 2.0, respectively. The corresponding loads on the inner rail are 33,000, 46,000, 45,000 and 36,000 lb. These loads show that there is a considerable increase in the bearing pressure on the inner rail over its normal value. For the higher speeds the stresses in the two rails are much the same for all the drivers. For the 6° curve the loads on the second and third drivers on the inner rail calculated from the stresses at 5 miles per hour (see Table 23) are 50,000 and 53,000 lb., respectively, and at 25 miles per hour 45,000 and 47,000 lb.

It is seen that for all the curves the relation of the average of the vertical bending stresses under all the wheels of a locomotive on the inner rail to the corresponding average of the stresses in the outer rail, taken as a measure of the bending of the rail in a vertical plane, in the main conform fairly closely to what may be expected from the analytical consideration of the transverse inclination of the track and the centrifugal force, so far as the locomotive as a whole is concerned. The principal variation from this conformity to analysis is the Pacific type locomotive on the 10° curve having a superelevation of 2 inches.

The outstanding result of the foregoing study of vertical bending stresses on curved track is the variable distribution of load among the drivers, and particularly the greatly increased loads found on some of the drivers beyond that normally expected. These high loads occur principally on the inner rail and are the most marked at the low speed, though high values were found in a number of cases at speeds two-thirds the way toward that of superelevation. Further examples of high loads under individual drivers at moderate speeds are cited in the discussion of the tests in California (see Article 20, Results of Tests on 10° Curves in California). Loads ranging up to 100 per cent in excess of the normal load were found with several of the locomotives. These excessive loads produce high stresses in the rail, but more important still, they give very high bearing pressures on the rail. It is a well-known fact, of course, that the inner rail of curves suffers greatly from the high pressures on the rail. The bearing pressures on the rail on straight track constitute a serious problem in the study of the rail, and the conditions on the curves seem to be far worse than have been thought. As the bearing pressures found are considerably higher under some of the wheels than those which ordinary analysis of the effect of transverse inclination of track and centrifugal force would predict, it would seem well to have a study made of the action of the equalizing levers and the springs on curved track to try to find the cause of the concentration of load, and to attempt to modify the design so as to obviate such gross inequalities.

15. **Lateral Bending of Rail on Curves.**—When forces are applied laterally to the rails of a curve by the wheels of the locomotive (the component forces in the direction of the radius of the curve, either inward or outward, being meant), lateral bending of the rail is produced and a set of stresses is developed, which may be termed lateral bending stresses.

As the beam strength of the rail in a lateral direction is much less than that which resists bending in a vertical direction (the section modulus, $\frac{I}{c}$ about a vertical axis being only about one-fifth of that about a horizontal axis), large lateral bending stresses will be developed by a lateral bending moment relatively much smaller than the vertical bending moment that causes vertical bending stresses of the same magnitude. The stress in any fiber will, of course, be the sum of the fiber stress due to vertical

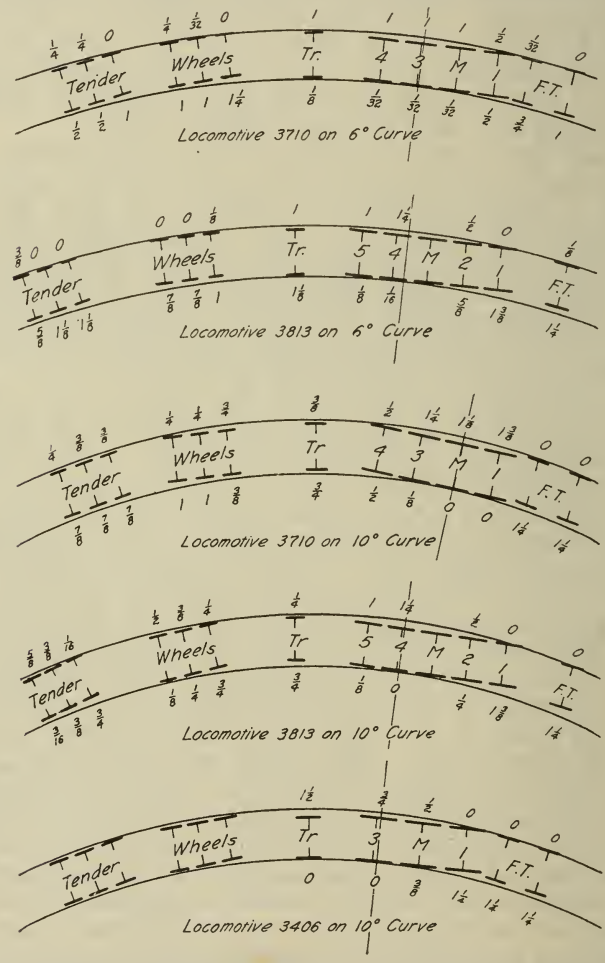


Fig. 73.—Relative Position of Wheels of Locomotives and Tenders with Respect to the Rails of Curved Track on the A. T. & S. F. Ry.

bending and that due to lateral bending, the nature of the stress either tension or compression being taken into consideration. It will help in understanding the cause of the lateral bending and in applying the analysis given in Article 12, The Action of Curved Track, if something is known of the position and direction taken by the wheels of the locomotive in traversing a curve.

In Fig. 73 and 74 are shown the relative positions of the wheels of five locomotives on several curves; the dimensions are given in inches and represent the distance from the gage side of the rail to the flange

of the wheel. The measurements were made with a view of obtaining a general notion of the position and direction taken by the wheels, but no effort was made to secure great accuracy or completeness in the tests and measurements. The results, however, may be taken to be fairly accurate. The locomotive was run over the track at slow speed and allowed to come to rest without the application of the brakes. The test was not repeated. On the A. T. & S. F. Ry. measurements were made between the gage side of the rail and the flange of the wheel. A steel scale was used and notwithstanding the difficulties of measurement the data are probably accurate to $\frac{1}{16}$ in. On the D. L. & W. R. R. no measurements were made except for the last diagram of Fig. 74, but observations were taken to see whether the flange bore on the side of the rail or was away

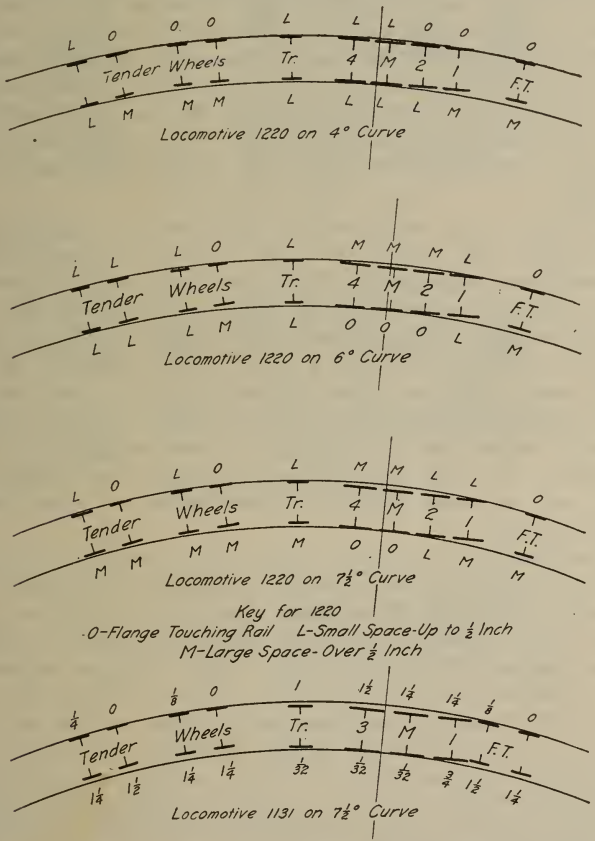

Tender Wheels Tr. 4 M 2 1 F.T.
Locomotive 1220 on 4° Curve
Tender Wheels Tr. 4 M 2 1 F.T.
Locomotive 1220 on 6° Curve
Tender Wheels Tr. 4 M 2 1 F.T.
Locomotive 1220 on 7½° Curve
Key for 1220
O-Flange Touching Rail L-Small Space-Up to ½ Inch
M-Large Space-Over ½ Inch
Tender Wheels Tr. 3 M 1 F.T.
Locomotive 1131 on 7½° Curve

Fig. 74.—Relative Position of Wheels of Locomotives and Tenders with Respect to the Rails of Curved Track on the D. L. & W. R. R.

from it a distance of less than one-half inch, or was more than one-half inch away. The radial line on the diagrams is drawn at the position where the radius of the curve is perpendicular to the line taken by the wheel base, the determination being made on a large scale drawing. The point where this radial line crosses the inner rail may be taken to be the probable center of rotation of the locomotive frame. Generally speaking, the flange of the outer front driver bears against or is close to the outer rail, and a line along the gage side of the flanges of the drivers cuts the

outer rail at the outer front driver or somewhat in front of it. The flange of the outer rear driver does not touch the outer rail and in several cases the flange of the inner rear driver is closer to the rail than that of the outer one. The flange of the inner front driver is well away from the rail. Generally, too, the flange of the driver next to the rear one is close to the inner rail, though it does not bear against it as does the outer front driver against the outer rail. The center of rotation (if this is the center of rotation) is the point where the radial line marked on the drawing crosses the inner rail. For the locomotives having eight or ten drivers this point is located on the inner rail at the driver next to the rear driver. Locomotive 3710 on the 10° curve appears to be an exception to this; the point is here found under the second driver. In the case of the Pacific type locomotive one diagram shows the point to be at the rear driver and one between the second and third driver. The former has the front driver flange against the outer rail; in the second one it is well away from it.

The flange of the outer wheel of the front truck generally bears against the outer rail. In the case of the four-wheel leading truck, both outer wheels have this bearing. The flanges of the trailers do not touch the outer rail. In one case only do they touch the inner rail. They take a position away from the two rails sometimes closer to one and sometimes to the other. The first wheel of the trucks of the tender generally bears against the outer rail and the line of the wheel base is a secant line cutting the outer rail at this point. In some cases the front truck does not take this position; its location seems to be somewhat variable. For those trucks having the front wheel bearing against the outer rail the center of rotation seems to be at or near the rear wheel of the truck.

The positions given in Fig. 73 and 74 may be taken to be representative of the position of the wheels at low speeds. For higher speeds some modifications may be expected. The tendency will be for the wheels to move somewhat towards the outer rail, though it seems evident that generally there will not be much change in position with respect to the two rails. There will be, of course, an increase of the pressure of the flange of the front wheel against the rail and also a change in the amount of lateral pressure transmitted by means of friction between the wheel and the rail, but these changes may not be expected to result in any marked change in relative position of the wheels and rails. This conclusion is borne out by the nature and amount of the lateral bending stresses found in the tests.

In Fig. 75 to 78 is recorded the magnitude of the lateral bending stresses in the outer and inner rails under the several wheels of the locomotive and tender at the speeds at which tests were made, the stress being given in pounds per square inch, for the tests at Ribera, New Mexico, Fort Madison, Iowa, and Dover, Paterson, and Mount Tabor, New Jersey. This lateral bending stress is the stress by which the stress

at one edge of the base of the rail exceeds that at the middle of the base of rail; its magnitude is found by taking one-half of the difference between the stresses at the inside and outside edges of the base of rail. The sign in front of the stress is indicative of the nature of the lateral bending of the rail; a positive sign indicates that the bending acts to increase the curvature of the rail, the negative that the bending

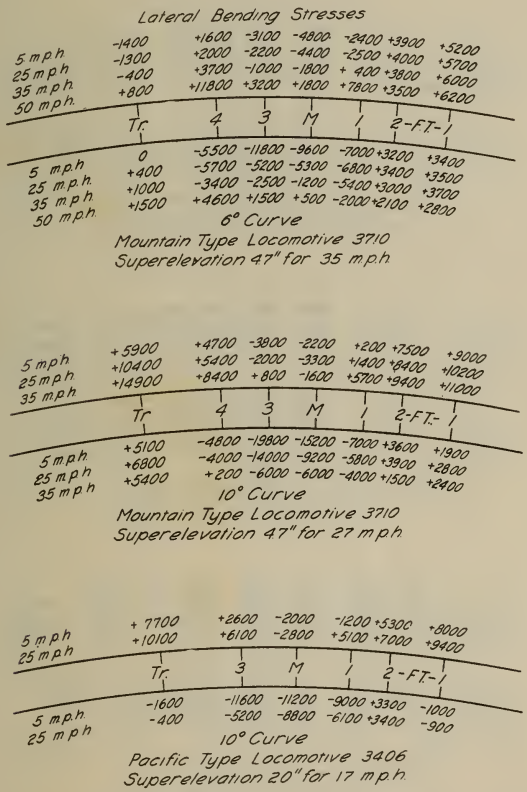
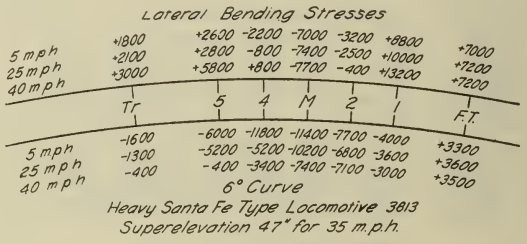


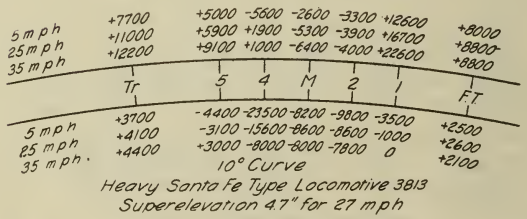
Fig. 75.—Lateral Bending Stresses in Base of Rails of Curved Track —Mountain and Pacific Type Locomotives of the A. T. & S. F. Ry.

tends to straighten the rail. The signs apply alike to both outer and inner rail. A positive bending in the outer rail at the point of contact of a wheel implies an outward thrust against the rail at the given point; a positive bending in the inner rail that a lateral pull on this rail acts toward the outer rail. An inspection of the diagrams will show that they give important information on the presence and nature of lateral

forces and the manner of the lateral bending in the rail. Some of the outstanding features may be noted as follows:

The lateral bending in the outer rail under the wheels of the front truck is found to be always positive (indicating an outward thrust) and it is generally positive also in the inner rail. This is true at all speeds. It indicates that the flanges of the outer wheel of the front truck





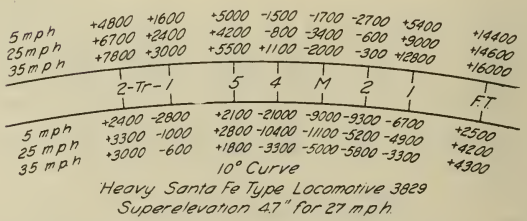


Fig. 76.—Lateral Bending Stresses in Base of Rails of Curved Track —Heavy Santa Fe Type Locomotives of the A. T. & S. F. Ry.

always have an important part in changing the direction of the locomotive.

In the locomotive having two-wheel front trucks the lateral bending of the outer rail at the outer front driver is positive at all speeds, with the exception of the Mikado type on the 6° curve, where the superelevation is 8.5 in. With the locomotives having four-wheel leading

trucks the bending at the outer front driver is generally negative at the low speed, becoming positive at a higher speed, with the exception that it remains negative in the case of the Pacific type locomotive on the 7½° curve. The nature of the bending of the outer rail is in agreement with what is to be expected from the position of the outer front driver given in Fig. 73 and 74.

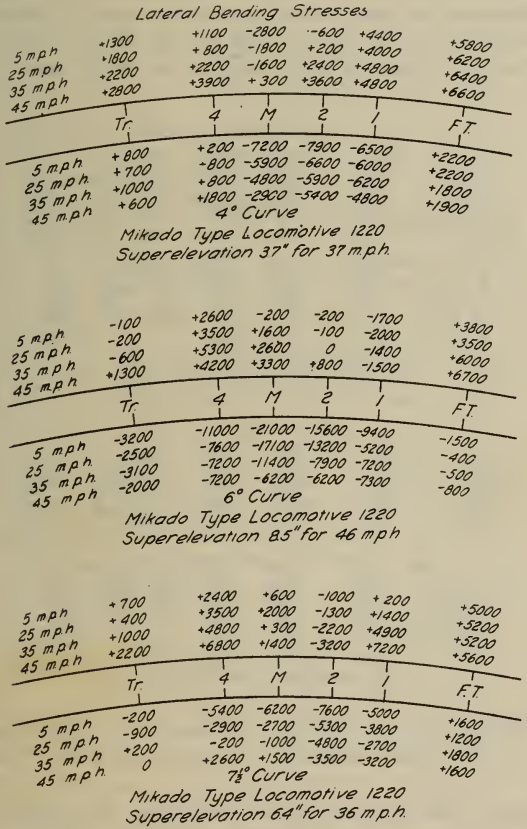


Fig. 77.—Lateral Bending Stresses in Base of Rails of Curved Track —Mikado Type Locomotive of the D. L. & W. R. R.

The outer rear driver gives a positive lateral bending in the outer rail (outward thrust on the rail) at all speeds, the one exception being the Pacific type locomotive on the 4° curve at a speed of 5 miles per hour.

At low speeds all the other drivers generally give negative bending in both inner and outer rail, tending to straighten the rail. The principal

exception is that the bending moment under the inner fifth driver of the Santa Fe type locomotive is sometimes positive.

An increase of speed results in an increase of lateral bending in the outer rail, decreasing the value of the lateral bending stress if the bending is negative and increasing the stress if it is positive.

An increase of speed results in a decrease in the lateral bending in the inner rail, the numerical value of the lateral stress decreasing when the bending moment is negative and increasing when it is positive.

The trailer has an outward thrust on the outer rail in every case at the higher speeds and almost generally at the lower speeds. The

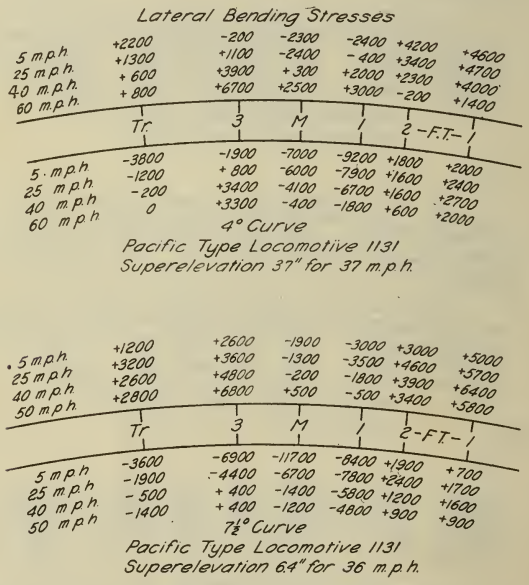


Fig. 78.—Lateral Bending Stresses in Base of Rails of Curved Track —Pacific Type Locomotive of the D. L. & W. R. R.

lateral bending of the inner rail under the trailer may be one way or the other.

For the wheels of the tender, a study of Fig. 42 to 63 will show that when there is positive lateral bending at the outer front wheel of a truck the position of the wheel in Fig. 73 and 74 shows its flange bearing against the outer rail. There is no uniformity, however, in the way in which the bending stresses under the trucks of the tender are distributed in the various locomotives.

The lateral bending stresses in the base of rail found on curved track and shown in these diagrams are of considerable magnitude—in

many cases extremely high. Severe lateral bending stresses were found in the inner rail of the 10° curve under the fourth driver of the Santa Fe type locomotive at the low speed. The value given on one of the diagrams is 23,500 lb. per sq. in. In the tests at Bealville, California, which will be described in Article 20, Results of Tests on 10° Curves in California, the lateral bending stress under the fourth driver of the Santa Fe type locomotive of the A. T. & S. F. Ry. at the low speed was 25,800 lb. per sq. in. and that under the same driver of the Southern Pacific was 20,300 lb. per sq. in. (see Fig. 96). Under the outer front driver of Locomotive 3813 at the highest speed the lateral bending stress in the outer rail was 22,600 lb. per sq. in. The lateral bending stress in the inner rail of the 10° curve under the third driver of the Mountain type locomotive at 5 miles per hour was 19,800 lb. per sq. in. and the similar stress on the 6° curve was 11,800 lb. per sq. in. (see Fig. 75). The lateral bending stress under the trailer of the Mountain type on the 10° curve at 35 miles per hour was 14,900 lb. per sq. in. The Pacific type locomotive developed a lateral bending stress of 11,600 lb. per sq. in. under the third driver in the inner rail of the 10° curve at Fort Madison at 5 miles per hour and of 10,100 lb. per sq. in. in the outer rail under the trailer at a speed of 25 miles per hour (see Fig. 75). The Mikado type locomotive developed a lateral bending stress of 21,000 lb. per sq. in. in the base of the inner rail of the 6° curve under the third driver and one of 15,600 lb. per sq. in. under the second driver; these were at slow speeds with the track having a superelevation of 8.5 in. (see Fig. 77). On the 7½° curve with a superelevation of 6.4 in. the corresponding lateral bending stresses were 6,200 and 7,600 lb. per sq. in., respectively (see Fig. 77). On the 4° curve the corresponding lateral bending stresses were 7,200 and 7,900 lb. per sq. in. The Pacific type locomotive on the 4° curve of the D. L. & W. R. R. developed 9,200 lb. per sq. in. lateral bending stress in the inner rail under the first driver and on the 7½° curve 11,700 lb. per sq. in. under the second driver; even on a 4° curve it is seen that the lateral bending stress under some of the wheels is considerable in comparison with the vertical bending stress in the rail under these same wheels (see Fig. 78).

When the stress due to lateral bending is combined with that due to vertical bending it will be seen that the resulting stress may be very great (see Fig. 42 to 67). This is the more marked because the high lateral bending stresses generally occur under wheels giving very high vertical bending stresses, much greater than those developed on straight track. Thus, for example, the stress at the outside edge of the base of the inner rail of the 10° curve under the fourth driver of the Santa Fe type locomotive at 5 miles per hour is the sum of 23,500 and 29,200 making a resulting stress here of 52,700 lb. per sq. in. This value is the average of a large number of runs; not infrequently stresses several thousand pounds greater than this were found—indeed one of

75,000 lb. per sq. in. was recorded. It may be added that the value of the lateral bending stresses under many of the wheels ranged from 40 to 100 per cent of the vertical bending stress under the same wheel.

The lateral bending stresses observed at points between wheels are of measurable magnitude. Their values bear somewhat the same relation to the stresses under the wheels as is found with the vertical bending

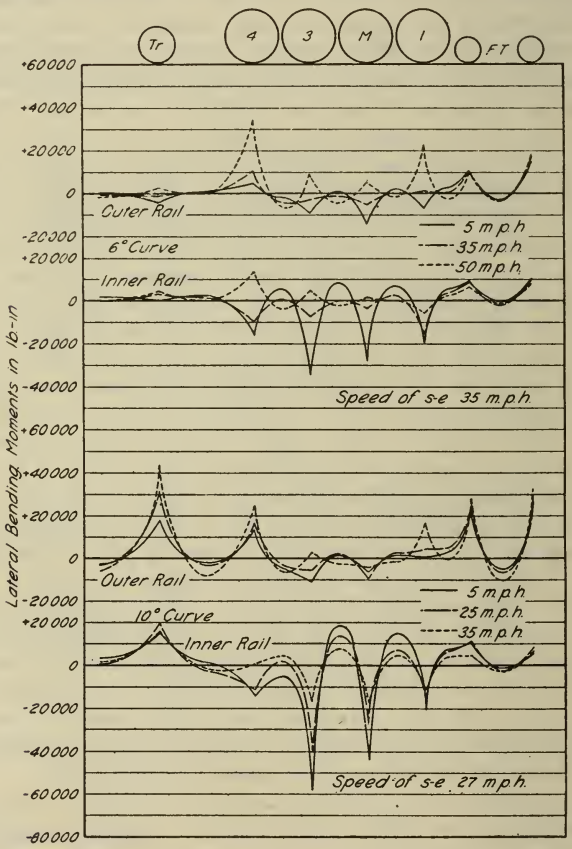


Fig. 79.—Lateral Bending Moments in Outer and Inner Rails of Curved Track with Mountain Type Locomotive of the A. T. & S. F. Ry.

stresses on straight track. In general, the direction of the bending at points between wheels is opposite in character to those at the wheels, positive bending if that at the wheels is negative and negative bending if the other is positive. This inward and outward bending corresponds

to the positive and negative bending moments found ordinarily at and between wheels for vertical load alone. The torsional stresses developed have not been analyzed.

16. **Lateral Bending Moments.**—The lateral resisting moment developed in the rails, which, of course, at any point is equal to the

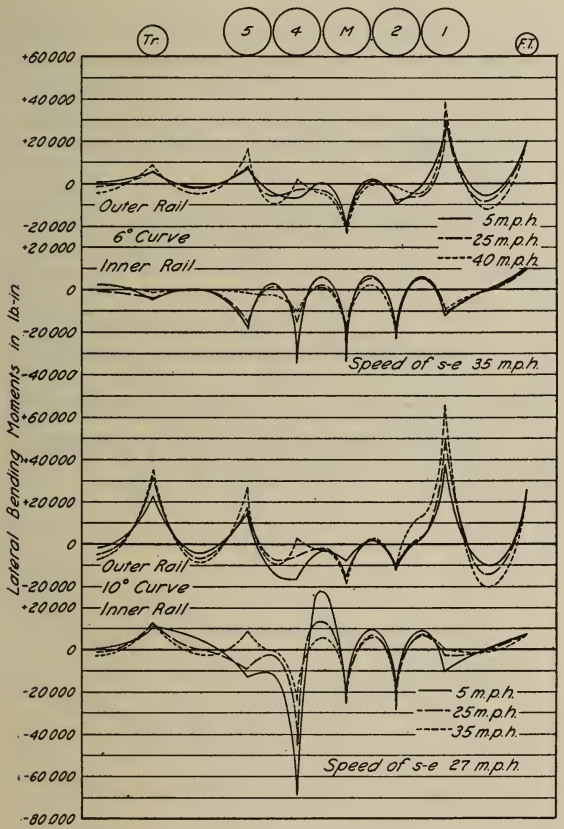


Fig. 80.—Lateral Bending Moments in Outer and Inner Rails of Curved Track with Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

lateral bending moment, may be readily calculated from the observed lateral bending stress by the use of the lateral section modulus $\frac{I}{c}$ of the rail. The variation in the lateral bending moment at and between wheels throughout the length of the locomotive and at the several speeds

offers an interesting view of the lateral bending action of the rails of curved track as the load passes. Fig. 79 to 83 give diagrams of the lateral bending moment in the outer and inner rails for four type of locomotives on several curves of different degrees of curvature. As

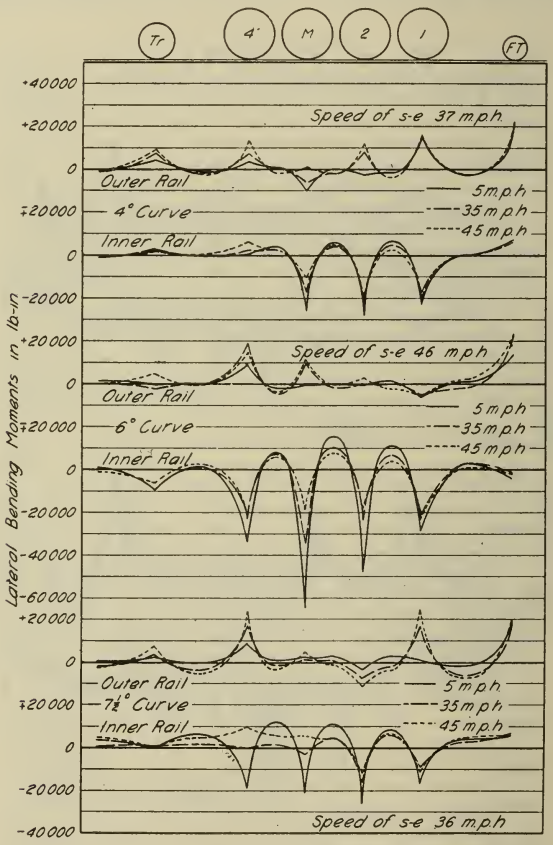


Fig. 81.—Lateral Bending Moments in Outer and Inner Rails of Curved Track with Mikado Type Locomotive of the D. L. & W. R. R.

before, positive bending is taken to be that which increases the curvature of the rail, and negative bending that which tends to straighten the rail.

A study of these diagrams brings out interesting characteristics of the different locomotives. The Mountain type on the 6° curve (Fig. 79) has much the same characteristics as on the 10° curve, though on the 6° curve except at low speed the lateral bending

moments are small with the exception of the outer rail under the fourth driver. Except at the speed of 50 miles per hour the principal turning action is given by the wheels of the front truck; on both curves at the high speed the front outer driver also participates in the turning action. With the Santa Fe type locomotive (Fig. 80) the magnitude of the bending

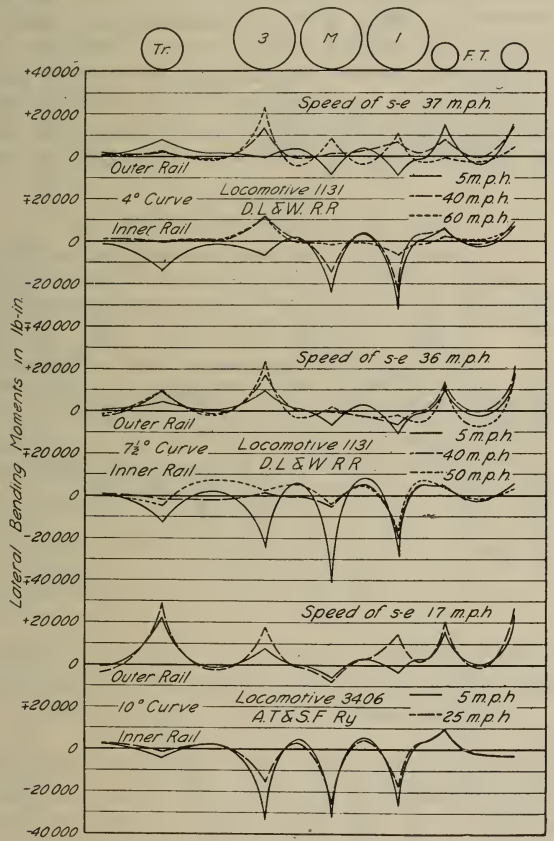


Fig. 82.—Lateral Bending Moments in Outer and Inner Rails of Curved Track with Pacific Type Locomotives

moment on the 10° curve is quite marked and it varies considerably with the speed. In this, as with the other locomotives, it will be noted that the lateral bending moment developed by the wheels of the front truck does not vary greatly with a change in speed, while that under the outer front driver generally increases materially with an increase in speed. It is apparent that the trailer has sufficient lateral swing on

the 6° curve to give freedom from lateral bending action upon the rail; on the 10° curve the lateral bending in the outer rail under the trailer is very noticeable. The Double Trailer Heavy Santa Fe type locomotive (Fig. 83) gives small lateral bending at points back of the fourth driver. It will be noted that the sum of the lateral bending moments in the outer rail under the front truck and first driver with this locomotive is equal to the corresponding sum found under the same wheels with the ordinary Santa Fe type locomotive. It will be remembered that the locomotive diagrams and the observed vertical bending stresses show that in the double trailer type a part of the load has been transferred from the first driver to the front truck wheel. The Pacific type locomotive on the 10° curve (see Fig. 82) shows that

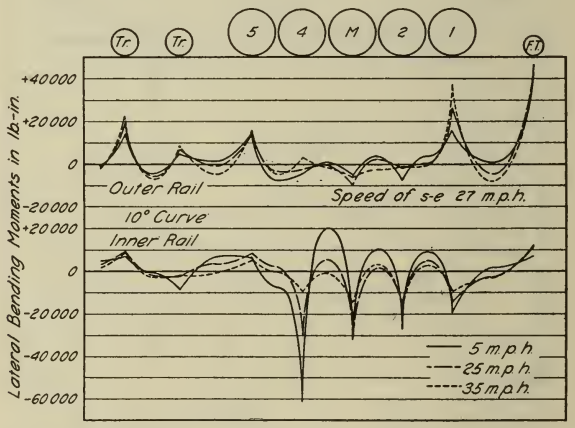


Fig. 83.—Lateral Bending Moments in Outer and Inner Rails of Curved Track with Double Trailer Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

the lateral bending moment in the inner rail may be considerable, even when the superelevation is small. It is evident also that the trailer of this locomotive does not have full freedom to swing, but bears against the outer rail as the locomotive is turned on the curve.

The Pacific type of the D. L. & W. R. R. (Fig. 82) shows a bending moment in the 105-lb. rail of only moderate magnitude except at a speed of 5 miles per hour on the 7½° curve. The bending moment in the inner rail under the second and third drivers of the Mikado type locomotive on the 6° curve at a speed of 5 miles per hour is much more marked than that on the 4° and 7½° curves. The stress due to vertical bending in the inner rail under the main driver is not much greater in the 6° curve than in the 7½° curve, so that it may be expected that the large amount of superelevation is not the only cause of the large bending moment.

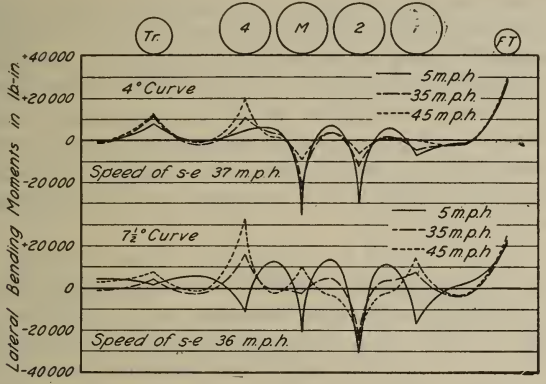


Fig. 84.—Combined Lateral Bending Moments in the Two Rails of Curved Track with the Mikado Type Locomotive of the D. L. & W. R. R.

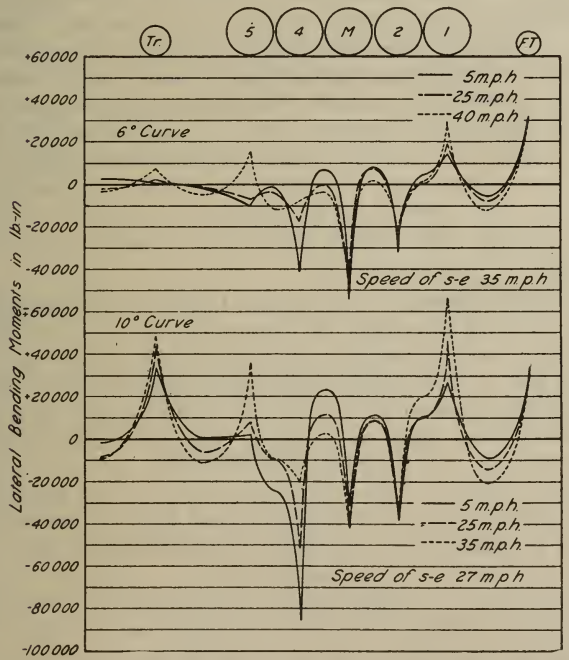


Fig. 85.—Combined Lateral Bending Moments in the Two Rails of Curved Track with the Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

The lateral bending moments for the outer and inner rails have been added algebraically and plotted in Fig. 84 and 85. The general trend of the combined bending moments under and between the several wheels is very striking. The diagrams in Fig. 79 to 83 brought out the large bending action developed in the two rails separately; these diagrams show the intense and severe lateral bending action which comes on the track as a whole as the locomotive passes around the curve, tending to push the ties and ballast first to one side and then to another and to throw the curve out of line.

It may be of interest to estimate this lateral bending moment in terms of the vertical bending moment of the rail under the same wheel. The ratio of the lateral bending moment in the inner rail under the fourth driver of three Santa Fe type locomotives (two of them belonging to the A. T. & S. F. Ry. and one to the Southern Pacific Company) to the vertical bending moments developed at the same time averages 0.16 at the lowest speed on the 10° curves. It should be recalled too that the vertical bending stress under this driver was far greater than the normal and that the pressure of the driver on the rail must have been nearly twice the normal load. The ratio of the lateral bending moment under the same wheels to the vertical bending moment found generally on straight track averaged 0.04, though values as great as 0.14 were found. As the section modulus of the rail about a vertical axis is only about one-fifth that about a horizontal axis, the lateral bending stresses in the rail are relatively great in comparison with the lateral bending moments. It will be seen that the lateral bending effect on the 10° curve under the fourth driver of this locomotive (taking the average value found) is at least four times the average value found on straight track and at least twice the highest value that may be expected.

17. **Measurements of the Lateral Deflection of the Rail and the Distortion of the Alinement of Curves.**—In order to learn to what extent the rail deflects laterally and the alinement of the curve changes under the load of a locomotive and to see the conditions under which lateral bending moments of considerable magnitude are developed, some measurements were made on both 6° and 10° curves. An instrument was devised with which the middle ordinate of a 6-ft. chord along the rail was measured. The two bearing ends of the instrument were placed in punch marks in the side of the head of the rail and the plunger of an Ames dial at the middle of the instrument was also placed in contact with the side of the head of the rail. The measurements were taken at every 3 ft. of length of rail on both inner and outer rail, thus overlapping one another. A set of readings (zero reading) with no load on the track was taken; the locomotive was then run on and a second set of readings taken, check readings also being generally made. The work of the tests was not as complete as would be liked; it, however, shows the nature of the movement of the rails. More tests of this nature should give information of value.

Fig. 86 shows the alinement of the track with the locomotive standing on the 6° and 10° curves at Ribera with reference to the original alinement. The values recorded on the diagrams represent the distance the rail is deflected from its original position, the measure-

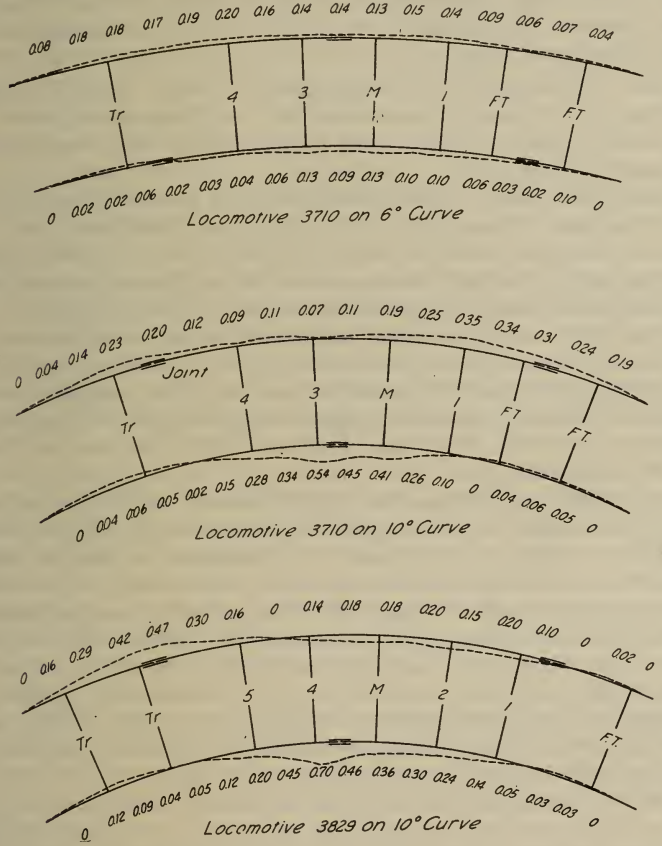


Fig. 86.—Changes in the Alinement of the Rails of Curved Track with the Locomotive Standing on the Curve.

ment being given in inches. As the measurements were taken on the head of the rail a part of the distortion of the alinement may be due to tilting of the rail on the tie. Necessarily in enlarging the small dimensions sufficiently to show the new position of the rails the lines on the diagram appear distorted and also the axles are not shown as parallel to each other. Attention is called to the position of the rail joints; their smaller lateral stiffness evidently contributes to the lateral deflection.

It is seen that with the Mountain type locomotive on the 10° curve the inner rail deflects more than one-half inch from its original position and that the gage of the track is increased from 4 ft. 8⅞ in. to as much as 4 ft. 9½ in. The outward deflection of the outer rail at the front truck wheel and first driver is large, 0.35 in. at the first driver. The greatest deflection is that of the inner rail under the third driver, 0.54 in. The changes from positive to negative bending between wheel points are evident.

With the Mountain type locomotive on the 6° curve the lateral deflection of both rails is much less, that of the inner rail under the third driver being only 0.13 in., and the gage of the track was increased only about ¼ in.

With the Double Trailer Santa Fe type locomotive on the 10° curve the inner rail under the fourth driver was deflected 0.7 in. The outer rail under the fourth driver was deflected inwardly of the track 0.14 in. so the gage of the track was increased 0.56 in. There was also 0.42 in. outward deflection of the outer rail at the first trailing wheel.

The general shape of the alinement which the rails take and the position of the many points of contraflexure are in general accord with the lateral bending moment diagram given in the preceding article. It is plain that the alinement of curves changes materially as the locomotive passes around it and that the track as a whole has large and varied stresses put into it.

18. **Effect of Speed and Counterbalance on Curves.**—It has already been found that the average of the stresses in the two rails of curved track under all the wheels of a locomotive at a speed of 5 miles per hour, found by dividing the sum of the stresses under all the wheels of the locomotive for both rails by the number of wheels in general agrees closely with the average of the corresponding stresses under the same wheels on straight track. If the average stress under all the wheels for the two rails of any curve be found for each of the speeds of any test and compared with the corresponding average stress on the straight track, it will be seen that the average stress on the curved track is generally no higher and sometimes lower than on straight track, or at any rate that the effect of speed on the average stress under the locomotive as a whole does not differ greatly on curves from that found on straight track, even though the transfer of weight from wheel to wheel on one side of the locomotive and from the wheel at one end of an axle to that at the other, found on curves, does change the stresses under individual wheels very greatly. The lateral bending stresses in rails on curves are, of course, not involved in the foregoing statement.

The effect of counterbalance upon the stresses in rail on curved track is much more variable than that found on straight track. In a few cases well-defined counterbalance curves could be drawn through the plotted points, but generally no line seemed representative of the effect of counterbalance. The points are scattered throughout a large vertical

range and there is much more variation in the results than is found on straight track. In estimating the stress that will include the effect of counterbalance the best that can be done is to take the average stress throughout the revolution of the driver and then allow for a very much greater variation in the stress above and below this line than would be done for straight track. One interesting result of the study of the data is that under a number of the wheels the stresses at the outside edge of the rail showed marked effect of counterbalance, while those at the inside edge show almost no counterbalance effect.

The effect of speed upon the division of the load of the locomotive between the two rails of the curve is quite another matter. In most cases the total load on the outer rail transmitted by all the wheels of one side of the locomotive bears the numerical relation to the total load on the inner rail that would be given by the analytical calculation of the effect of transverse inclination of track and centrifugal force at the given speed. This means that at slow speeds the load on the inner rail is considerably greater than the normal load, the vertical bending stresses in the inner rail with the Mikado type locomotive running at a speed of 5 miles per hour being found to be as much as 37 per cent greater with the superelevation of 8.5 in. on the 6° curve of the D. L. & W. R. R. than were found on straight track. At and near the speed of superelevation, in most of the tests on curved track, the vertical bending stresses in the two rails do not differ much from each other. It may be noted that analysis shows that the speed may be increased considerably beyond the speed of superelevation before the load on the outer rail would become equal to that imposed on the inner rail at low speeds. While the ratio varies with the degree of curve and the superelevation used it may be said that for the track used in the tests the speed may be increased 30 to 40 per cent beyond that of superelevation before the load on the outer rail may be expected to equal that found on the inner rail at low speeds. The foregoing discussion refers to the sum of the loads under all the wheels; the distribution of load among the several drivers may be quite unequal, the vertical bending stresses in the rail under one driver of one type of locomotive being as much as double what might be expected in the case of the 10° curve and 50 per cent more on the 6° curve.

The lateral bending stresses developed in the rails also vary with the speed. The lateral bending stresses in the inner rails are very great at the slow speeds, especially at the driver in front of the rear driver, and sometimes the one in front of this, in the case of the locomotives having eight and ten drivers. For medium speeds, say up to two-thirds of the speed of superelevation, the lateral bending stress under the rear intermediate driver continues to be of considerable magnitude. For the higher speeds the lateral bending stresses under these wheels decrease markedly and at the speed of superelevation the highest lateral bending stress in the inner rail may not differ greatly from the highest found in

the outer rail. The greatest lateral bending stress in the outer rail under a driver is found at the outer front driver in those locomotives in which the wheel of the leading truck does not assume the principal part in changing the direction of the locomotive. With an increase in speed this lateral bending stress increases. As the stresses in the outer rail under the wheels of the front truck do not increase with an increase in speed it is evident that the greater effort to change the direction of the locomotive at the higher speeds is taken by the flange of this first outer driver.

The effect of speed on the stress at the two edges of the base of rail (combined vertical bending stress and lateral bending stress) is well shown in Fig. 64, 65, 66 and 67. A marked decrease in the stress at the outside edge of the base of the inner rail under the driver in front of the rear driver with increase of speed is very noticeable. The increase of stress in both edges of the outer rail under the companion driver is also apparent. It may be noted also that the stresses at the four edges of the two rails under the wheels of the tender generally approach uniformity of magnitude at the speed of superelevation.

19. **Effect of Degree of Curve and Superelation.**—It has been seen that both the vertical bending stress and the lateral bending stress have wide ranges in value in the two rails and for the individual wheels of the several locomotives used. It may be expected that these differences will vary with the degree of curve and the superelevation of the outer rail, though it may not be possible to distinguish between the effects of the two. In comparing the action of the locomotives it will be convenient to refer to the values given in Fig. 69 to 72 and Fig. 75 to 78.

With the Santa Fe type locomotive on the 10° curve at 5 miles per hour the vertical bending stress of 29,200 lb. per sq. in. in the inner rail under the fourth driver may be contrasted with the stress of 19,400 lb. per sq. in. on the 6° curve having the same superelevation, 4.7 in.; the corresponding stress on straight track being about 12,000 lb. per sq. in. It is evident that vertical bending stress under this driver, which must be considered an excessive stress for both curves, increases with the increase in curvature although the curves have the same superelevation. At the speed corresponding to the superelevation the stress under this wheel in both curves decreased nearly to what might be expected under an even division of load among the drivers. The vertical bending stress in the inner rail of the 10° curve under the first driver at a speed of 5 miles per hour is less than would be expected and decreases with an increase of speed, while on the 6° curve it remains nearly constant for the three speeds. The vertical bending stress in the outer rail of the 10° curve under the first driver at 5 miles per hour is not much above what would normally be expected. The high value of 27,000 lb. per sq. in. at 35 miles per hour is, of course, at a speed well above that of superelevation. The vertical bending stresses in the outer

rail under the third and fourth drivers are small; that under the fourth driver at 5 miles per hour is very small, showing that the load transmitted by this driver is but a small proportion of its normal load; the driver opposite it on the inner rail takes most of the load. Even at the highest speeds the stresses in the outer rail of both curves under this driver are below normal values. It is evident that on both curves the distribution of the load among the drivers given by the equalizing system is markedly different from that on straight track. The lateral bending stress in the inner rail at 5 miles per hour is very great under the fourth driver, 23,500 lb. per sq. in. on the 10° curve and 11,800 lb. per sq. in. on the 6° curve, very severe stresses in both cases. Even at 25 miles per hour the stress recorded for the 10° curve was 15,600 lb. per sq. in. At the highest speed the lateral bending stresses in the inner rail become much smaller. The lateral bending stresses in the outer rail under the first driver, indicative of the participation of this driver in changing the direction of the locomotive at a speed of 5 miles per hour, are 12,600 lb. per sq. in. on the 10° curve and 8,800 lb. per sq. in. on the 6° curve. These lateral bending stresses increase markedly from the low speed to the highest, becoming 22,600 lb. per sq. in. on the 10° curve at 35 miles per hour and 13,200 lb. per sq. in. on the 6° curve at 40 miles per hour. Bearing in mind that the lateral bending stresses at the outer wheel of the front truck vary little with changes in speed, the results indicate that altogether a larger amount of lateral pressure is required to change the direction of the locomotive at the higher speeds than at the low speeds. The amount of the lateral pressure given by the front driver also increases somewhat with the degree of curve, though the influence of the curvature seems not as great as that of speed.

With the Mountain type locomotive on the 10° curve at 5 miles per hour the greatest vertical bending stress in the inner rail (average stress) is under the third driver, 26,700 lb. per sq. in. On the 6° curve the vertical bending stresses are high under the first, second, and third drivers, that under the third driver, 19,700 lb. per sq. in., being about the same as that under the other two. An increase of speed to that corresponding to the superelevation brings the vertical bending stresses in the inner rail of two curves under the fourth drivers down to about the same average value as the others, though that under the third driver on the 10° curve, 20,600 lb. per sq. in., remains greater than the others. At speeds above the speed of superelevation the vertical bending stresses in the inner rail have decreased materially and are much the same on both curves for all the drivers. The vertical bending stresses in the outer rail, particularly under the second and third drivers, are somewhat smaller on the 10° curve than on the 6° curve, these wheels being on the same axle as those giving very high vertical bending stresses in the inner rail. The stresses normally to be expected are not found until a speed higher than that of superelevation is reached. The lateral bending stress in the inner rail is greatest under the third driver at 5 miles per hour, 19,800 lb. per

sq. in. on the 10° curve and 11,800 lb. per sq. in. on the 6° curve. These values are roughly proportional to the degree of curve. On the 6° curve a small value is found at the speed of superelevation, but on the 10° curve even at 25 miles per hour the lateral bending stress is 14,000 lb. per sq. in., though the stress decreases rapidly with further increases of speed. In the outer rail, due probably to this locomotive having a four-wheel leading truck, the first driver does not participate in changing the direction of the locomotive except at the higher speeds. At 50 miles per hour on the 6° curve the outward lateral bending stress under the first driver becomes 7,800 lb. per sq. in. It is worth noting also that the outward bending stress in the outer rail under the fourth driver on the 6° curve reaches 11,800 lb. per sq. in. at a speed of 50 miles per hour. At the lower speeds it is evident that there is a transfer of load from the outer to the inner rail on both curves, but the change is the greater on the 10° curve. It is apparent that the lateral bending stress in the inner rail for low speeds is a function of the curvature, the superelevation being the same.

With the Mikado type locomotive on the D. L. & W. R. R. the even distribution of vertical bending stresses in the inner rail under all the drivers at a speed of 5 miles per hour is very apparent for all three curves, but the stresses differ on the three curves and it may be a question as to what extent these differences in stress are due to superelevation and to what extent to curvature. The vertical bending stresses under the four drivers average about 13,000 lb. per sq. in. on the 4° curve (3.7 in. superelevation, 105-lb. rail), 18,000 lb. per sq. in. on the 7½° curve (6.4 in. superelevation, 105-lb. rail), and 22,400 lb. per sq. in. on the 6° curve (8.5 in. superelevation, 92-lb. frictionless rail). It may be noted that this uniformity of stress under all the drivers is quite different from that found with the locomotives on the track of the A. T. & S. F. Ry. The vertical bending stresses in the outer rail have small values under all the drivers, particularly under the main driver. They do not reach the values found in the inner rail until a speed well above that corresponding to superelevation has been reached. The lateral bending stresses in the inner rail are much more marked in the 6° curve and are somewhat higher in the 4° curve than in the 7½° curve. It is not clear that the superelevation is the cause of this variation. It is worth noting that although this locomotive has a two-wheel leading truck, the lateral bending stresses in the outer rail under the first driver at a speed of 5 miles per hour show that this driver does not participate in changing the direction of the locomotive except on the 4° curve. On the 6° curve, which has the frictionless rail, the lateral bending moments in the outer rail at the first driver are negative for all speeds.

With the Pacific type locomotive of the A. T. & S. F. Ry. on the 10° curve at Fort Madison, Iowa, there being only 2 in. of superelevation, the vertical bending stress in the inner rail under the three drivers at a speed of 5 miles per hour is nearly double that in the outer rail. This goes to show that the curvature is here the main cause of shifting

the load of the locomotive to the inner rail, and not the superelevation. With the Pacific type locomotive of the D. L. & W. R. R. the shift of load to the inner rail is somewhat greater on the 7½° curve than on the 4° curve; both the curvature and the superelevation may be responsible for this. Considering the difference in the rail sections, the lateral bending stress in the inner rail at 5 miles per hour is really greater on the 7½° curve than that found on the 10° curve of the A. T. & S. F. Ry.; it is lower still on the 4° curve. The outer rail shows no turning action by the front driver on any of the three curves at a speed of 5 miles per hour, most of the change of direction evidently being accomplished by the wheels of the front truck. At the higher speeds the front driver participates in the turning action in the case of the 4° curve and the 10° curve, but not the 7½° curve.

To sum up the foregoing: On all the curves and for locomotives having eight and ten drivers, at the lower speeds the vertical bending stress is markedly greater in the inner rail under the next to the last driver than that in either rail under any other driver; for locomotives of the Pacific type the stress in the inner rail is greater but to a somewhat less degree under all three drivers than that in the outer rail. The vertical bending stress in the outer rail is correspondingly small under the companion driver of the one giving the high stresses in the inner rail. These effects are especially noticeable at the low speeds, they continue at moderate speeds, and they generally disappear at or above the speed of superelevation. Although part of the increased value of the vertical bending stress in the inner rail is due to the transfer of load to the lower rail by reason of the transverse inclination of the track, the excess over what may be expected on straight track is greater than may be accounted for by the superelevation, and frequently this discrepancy is unexpectedly great. It appears then that the equalizing system acts differently on curves than on straight track and that much more load is concentrated on this inner driver, the outer driver being relieved of a corresponding amount of load. This phenomenon was found on all the curves. Although this lack of even distribution of stresses under the drivers of one side was not present in the tests with the Pacific type locomotive, yet the vertical bending stresses in the inner rail with this locomotive are higher than may be explained by the transverse inclination of the track. It will be seen that on all the curves an increase of speed to one at or above the speed corresponding to the superelevation results in vertical bending stresses under the different wheels of the two rails more nearly what may be expected when the effect of transverse inclination and centrifugal force are considered.

The lateral bending stress in the inner rail under the driver giving the large vertical bending stresses and also under the one ahead of it is quite marked at the lower speeds on all the curves. Although the value of the stress is considerably greater on the 10° curve its magnitude is of considerable importance on the 6° and 4° curves.

The lateral bending stress in the outer rail under the first driver, resulting from the flange of this driver assisting in the change of direction of the locomotive, when found at the low speed, seems not to increase much with increased curvature of the track. As has already been noted this lateral bending stress does increase with an increase in speed.

20. **Results of Tests on 10° Curves in California.**—The data of tests made by the Atchison, Topeka and Santa Fe Railway and the Southern Pacific Company in April, 1922, on the tracks of the two railways, as they relate to the purposes of this report, have been made available to the Joint Committee. These tests were conducted on a 10° curve of the S. P. Co. at Bealville, California, and on a 10° curve of the A. T. & S. F. Ry. at Cajon, California. The locomotives were the Heavy Santa Fe type of the A. T. & S. F. Ry. and the Heavy Santa Fe type of the S. P. Co., both of them having oil burners. The rail at Bealville was 90-lb. A.R.A.-A. section; that at Cajon was 90-lb. S. F. section. As already noted, the properties of these rails are nearly identical. At Bealville there were 5 inches of new rock ballast on a thick mixed rock ballast foundation. At Cajon there were 10 inches of gravel ballast. The former gave the stiffer track.

The purpose of the tests was to determine the action and rigidity of the locomotives used in the tests. On the tests made at Cajon, a flangeless tire was substituted for the flanged wheel ordinarily used for the fourth driver, the third driver being flangeless as usual. As used at the first part of the tests, the bearing part of the tread of the fourth driver was cylindrical. Later a groove to approximate the conditions after wear was machined into the tread. Speeds of 10 to 18 miles per hour were used at both locations. The runs were made on 2.2 per cent compensated grade, net grades of 1.8 per cent on the A. T. & S. F. Ry. and of 2.0 per cent on the S. P. Co. Runs were made in two directions—up-grade and down-grade. The up-grade tests were made with the locomotive working steam and pulling from 9 to 12 loaded cars, the load pulled being altogether approximately 500 tons. The drawbar pull as measured by the dynamometer car placed immediately back of the tender recorded about 30,000 lb. on the various runs. On the down-grade tests the locomotives ran light and without working steam—coasting.

Fig. 87 to 94 give the average stresses for the several runs at the inside edge and outside edge of both inner and outer rails of the 10° curves at Bealville and Cajon, and at the two speeds. Fig. 95 and 96 give the values of the vertical bending stress and the lateral bending stress at Bealville and Fig. 97 and 98 the values at Cajon. Attention should be called to the fact that in the down-grade tests at Bealville with both locomotives it was raining and misting all day and the rails were continually wet; this seems to have had an effect on the stresses developed under one of the locomotives and to a less extent on the other.

The stresses in the rails with the locomotive of the A. T. & S. F. Ry. at Bealville are, in general, very much the same in magnitude and character

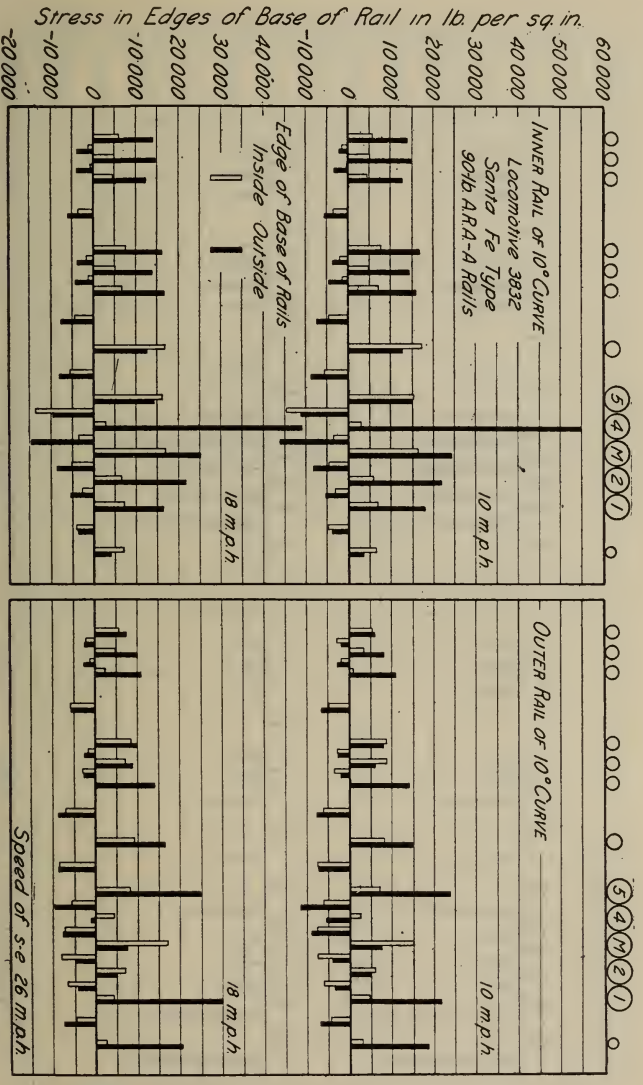


Fig. 87.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Bealville, California—Series 5500-03, Heavy Santa Fe Type Locomotive of the A. T. & S F. Ry. Pulling 540 Tons Up-grade.

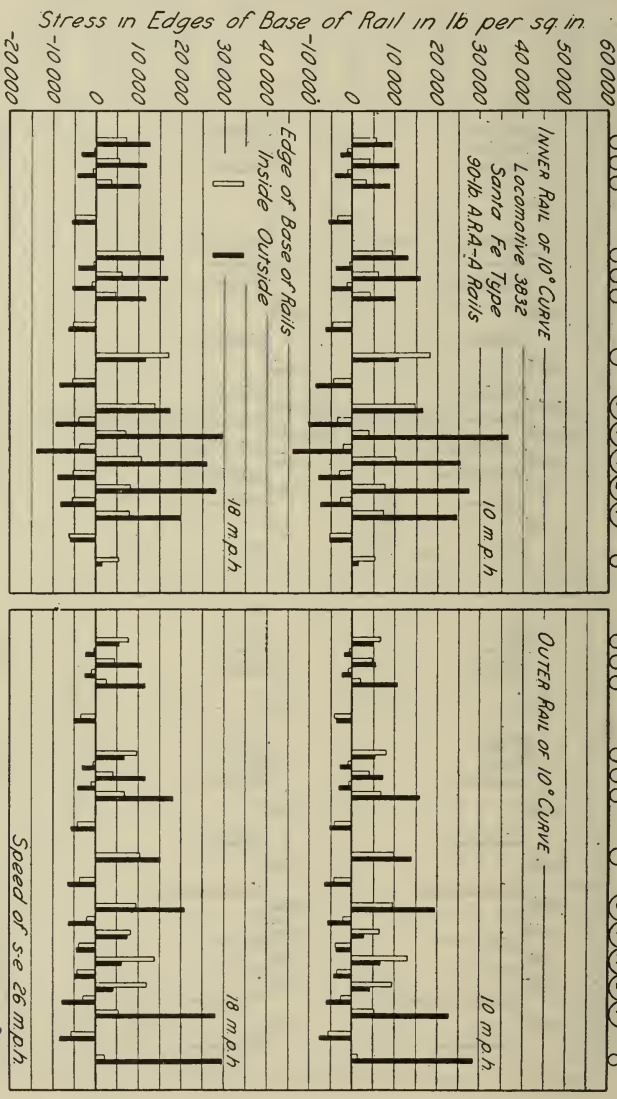


Fig. 88.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Bealville, California—Series 5504-07, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry. Down-grade, Light.

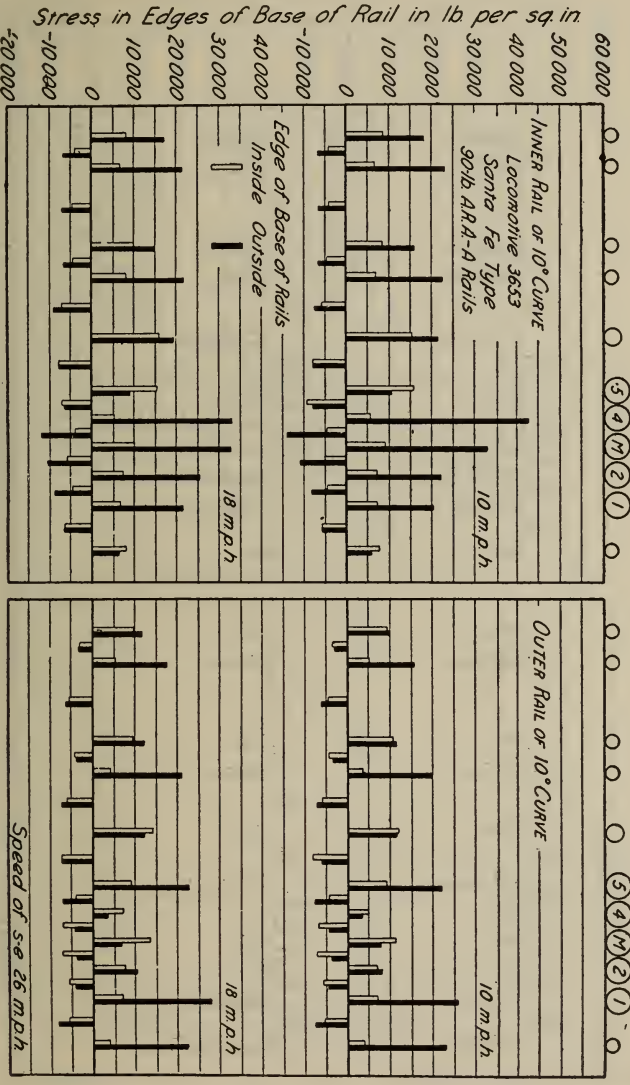


Fig. 89.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Bealville, California—Series 5513-16, Heavy Santa Fe Type Locomotive of the S. P. Co., Pulling 540 Tons Up-grade.

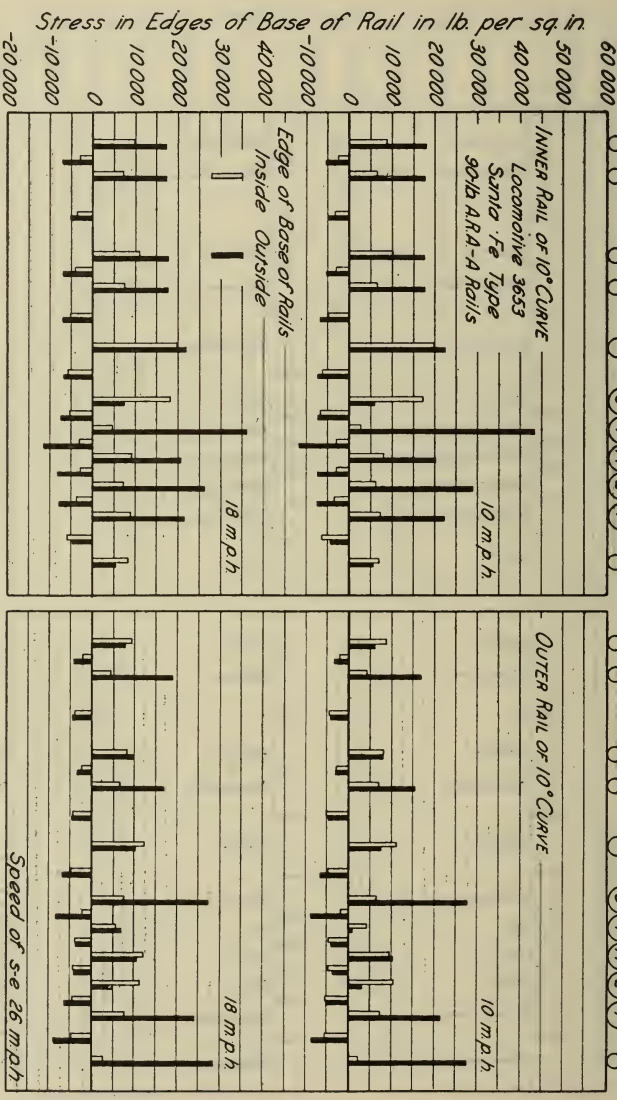


Fig. 90.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Bealville, California—Series 5508-12. Heavy Santa Fe Type Locomotive of the S. P. Co., Down-grade, Light.

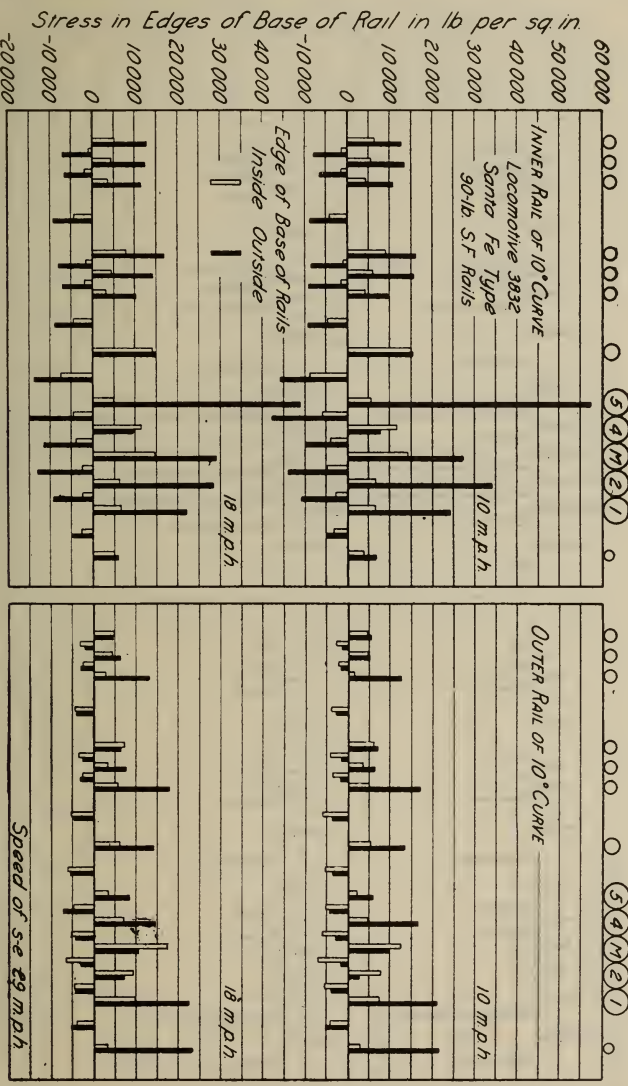


Fig. 91.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Cajon, California—Series 5523-26, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry. with 4th Driver Flangless and Tread Cylindrical. Pulling 500 Tons Up-grade.

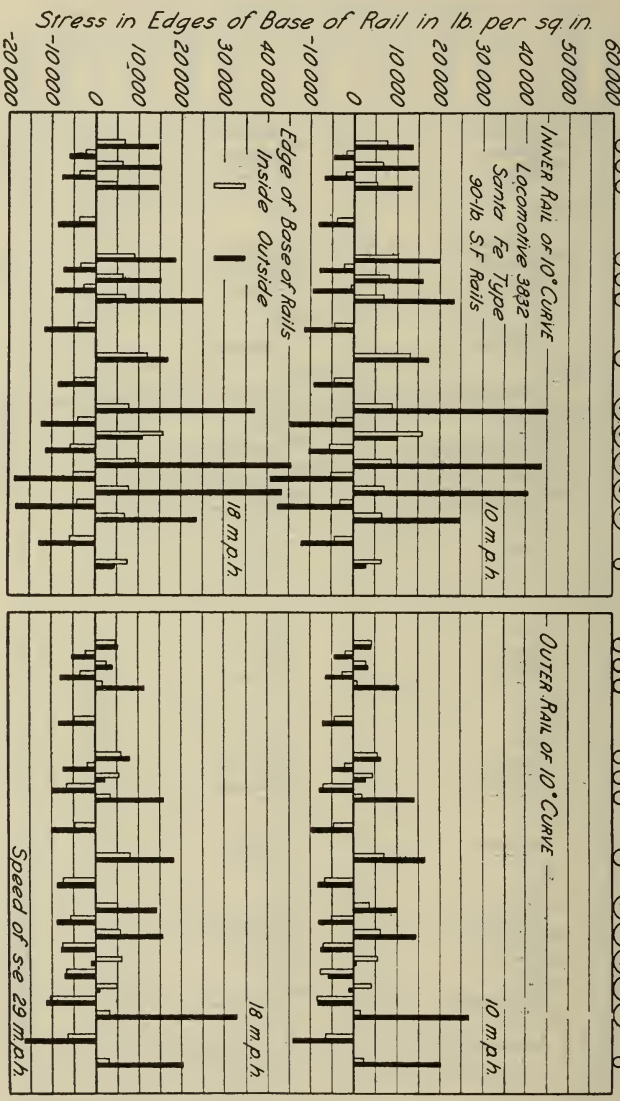


Fig. 92.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Cajon, California—Series 5519-22, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry. with 4th Driver Flangeless and Tread Cylindrical. Down-grade, Light.

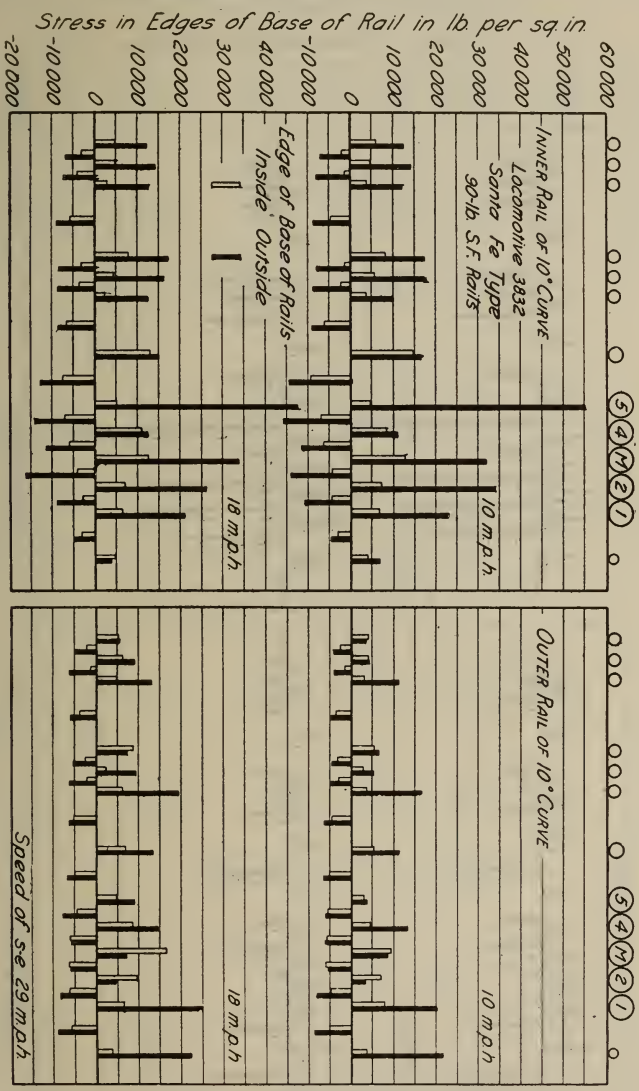


Fig. 93.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Cajon, California—Series 5527-30, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry. with 4th Driver Flangless and Tread Grooved. Pulling 500 Tons Up-grade.

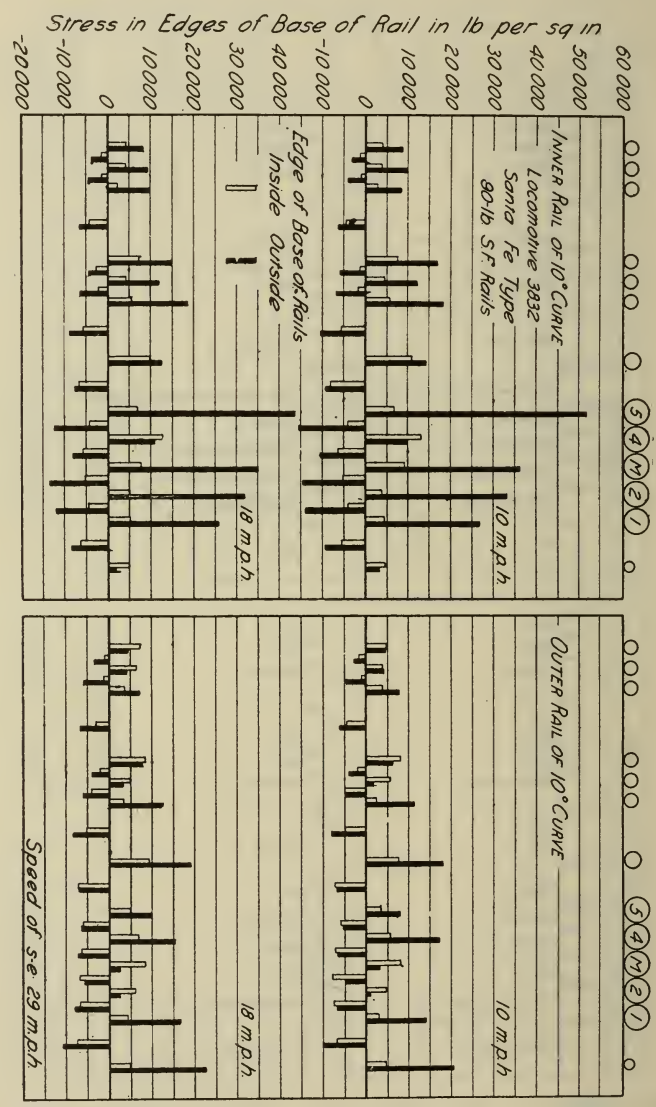


Fig. 94.—Stress at the Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Cajon, California—Series 5531-34, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry. with 4th Driver Flangeless and Tread Grooved. Down-grade, Light.

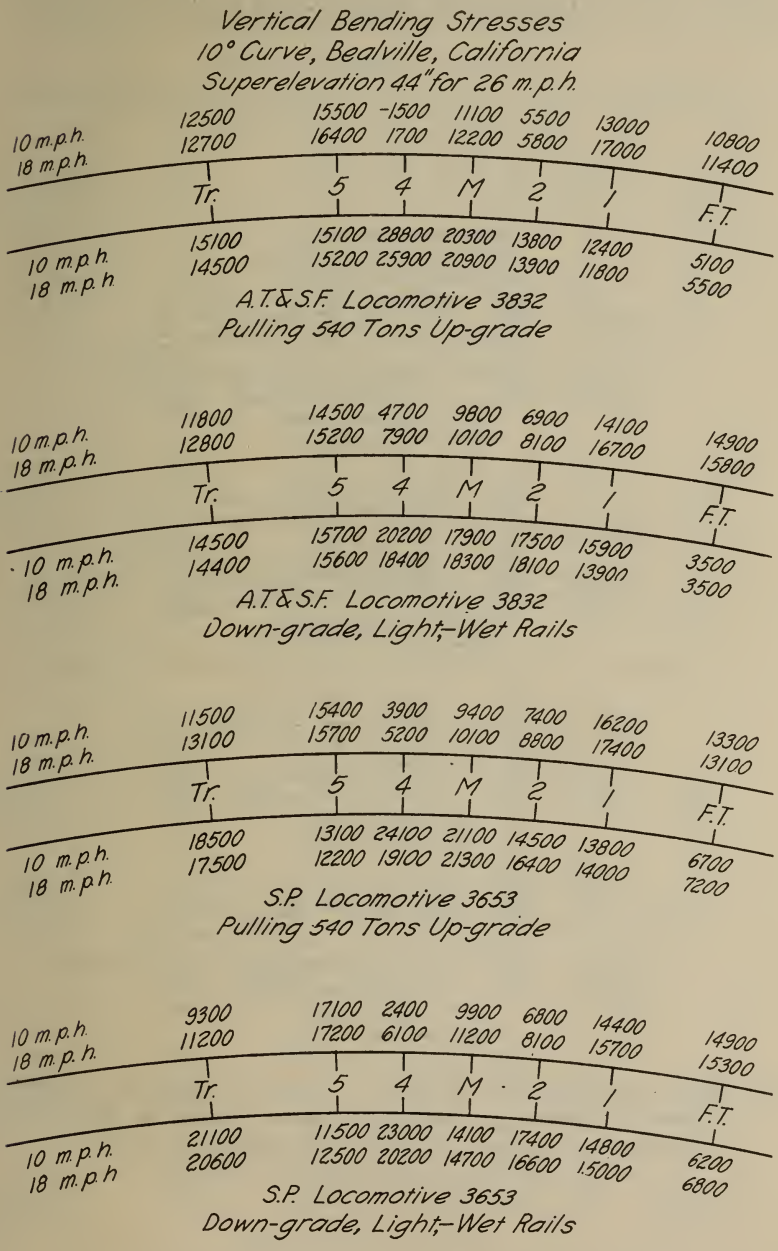


Fig. 95.—Vertical Bending Stresses in Base of Rails of the 10° Curve at Bealville, California—Heavy Santa Fe Type Locomotives of the A. T. & S. F. Ry. and the S. P. Co.

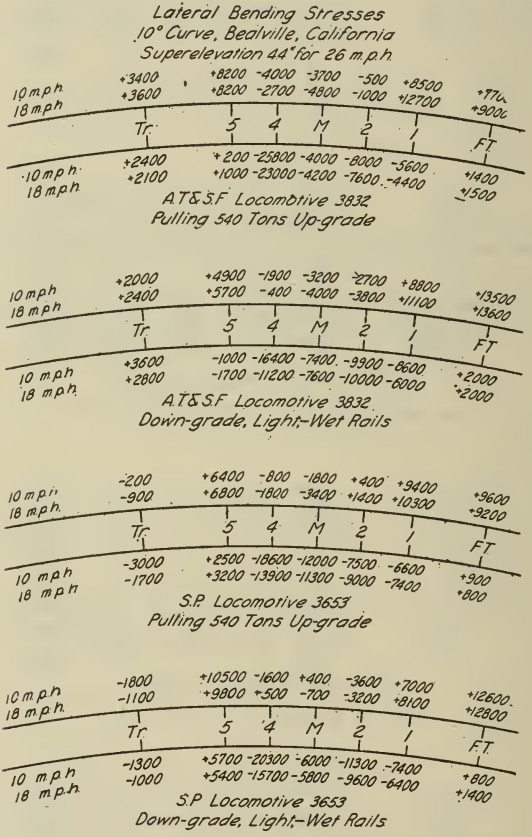


Fig. 96.—Lateral Bending Stresses in Base of Rails of the 10° Curve at Bealville, California—Heavy Santa Fe Type Locomotives of the A. T. & S. F. Ry. and the S. P. Co.

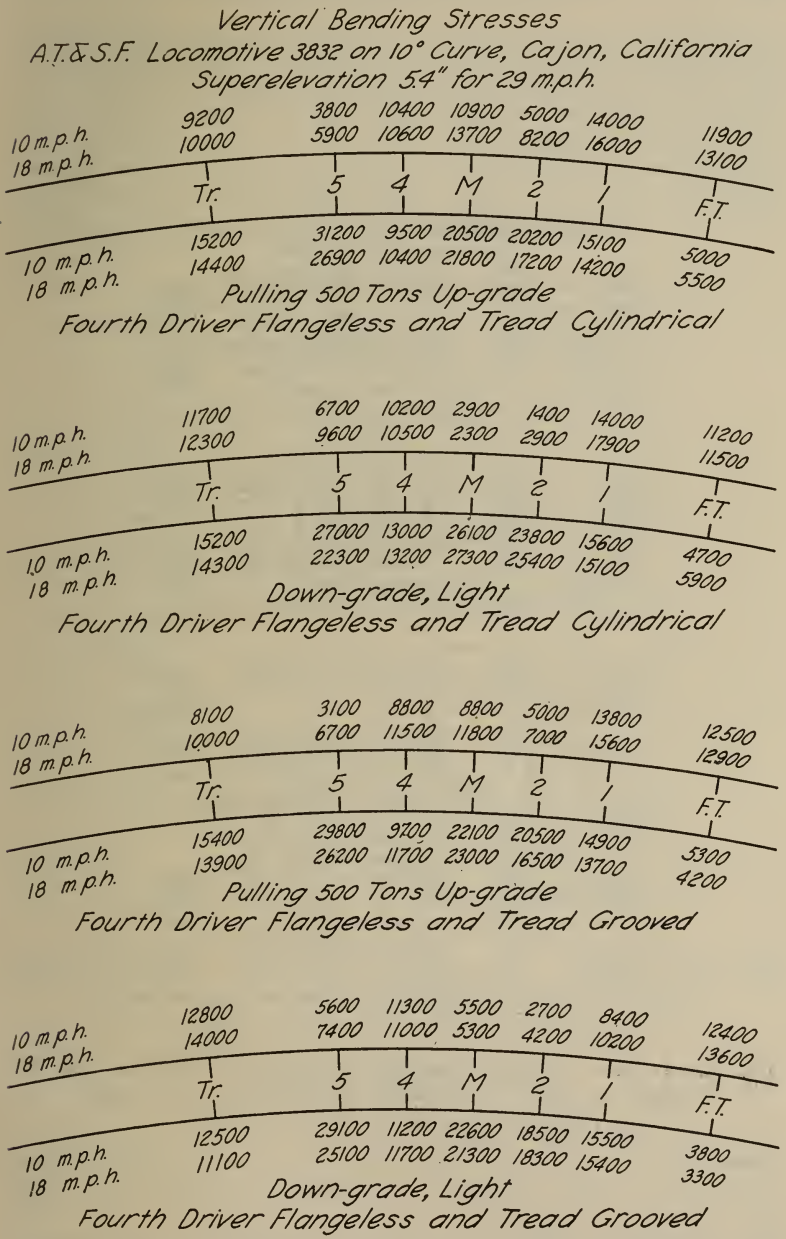


Fig. 97.—Vertical Bending Stresses in Base of Rails of the 10° Curve at Cajon, California—Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

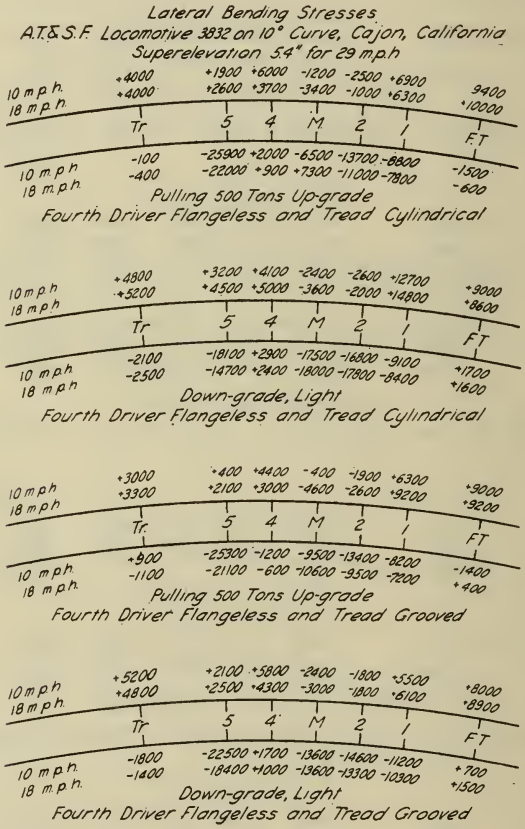


Fig. 98.—Lateral Bending Stresses in Base of Rails of the 10° Curve at Cajon, California—Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry.

as those found with a locomotive of the same class on the 10° curve at Ribera, New Mexico, except that on account of the locomotive used in California being equipped with oil burners the weight on the trailers was only about three-fourths of that of the locomotive used in New Mexico, and the stresses under the trailers were correspondingly smaller. The stresses with the locomotive of the S. P. Co. differed in some important respects from those under that of the A. T. & S. F. Ry., as did those with the locomotive with modified drivers used at Cajon.

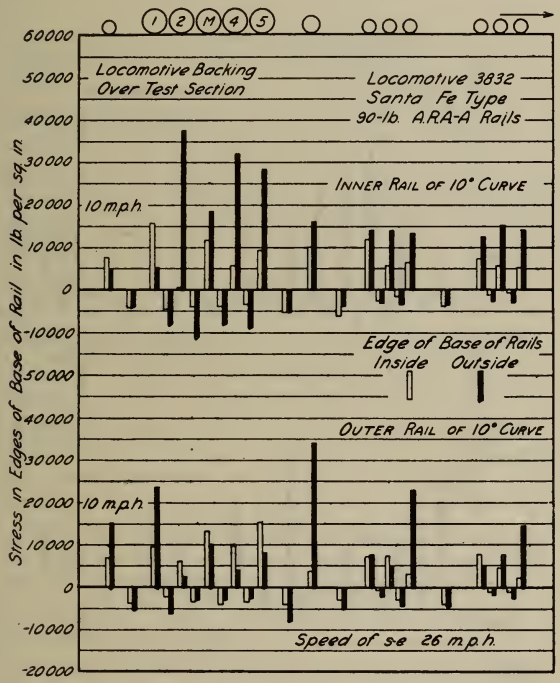


Fig. 99.—Stress at Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Bealville, California—Series 5518, Heavy Santa Fe Type Locomotive of the A. T. & S. F. Ry. Backing Down-grade, Light.

Table 23 shows that loads at individual wheels calculated from the vertical bending stresses at a speed of 10 miles per hour are excessively high, 50,000, 52,000, and 60,000 lb. These high loads do not necessarily occur under the same drivers, the variations seemingly being dependent upon the presence or absence of flanges. It will be seen, from Fig. 95 and 97 that the vertical bending stresses under the inner

driver that gives the high vertical bending stress and therefore transmits the high load are nearly as great at 18 miles per hour as at 10 miles per hour, the speed of superelevation being 26 miles per hour.

Fig. 99 and 100 give the average stress for the several runs at the inside and outside edges of both inner and outer rail of the 10° curve at Bealville when the locomotives were backing down-grade at 10 miles per hour. The rails were dry. The records of the locomotives of the two companies are given.

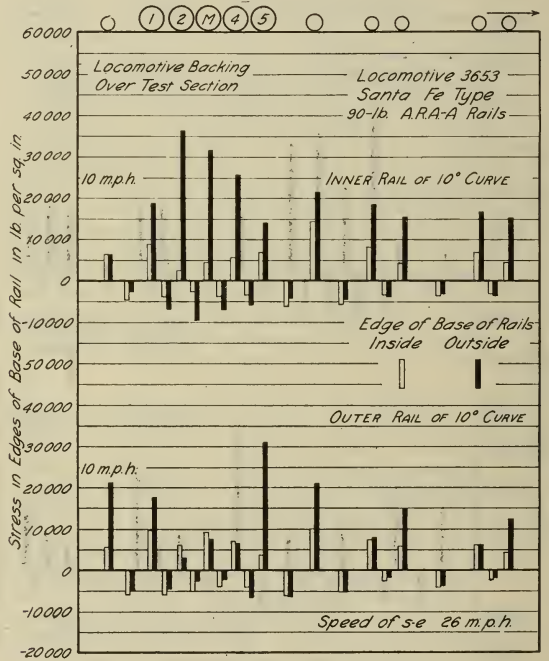


Fig. 100.—Stress at Inside and Outside Edges of the Base of the Inner and Outer Rails of the 10° Curve at Bealville, California—Series 5517, Heavy Santa Fe Type Locomotive of the S. P. Co. Backing Down-grade, Light.

The results of the tests on the 10° curves in California will be discussed with those of the other tests in the articles which follow on Type of Locomotive, Effect of Pulling and Coasting and Condition of Rail, and Tests With Locomotives Backing Over Curved Track.

21. **Type of Locomotive.**—On straight track the stresses developed in the rail are controlled by such matters as the wheel loads and

the wheel spacing. Lateral bending stresses are developed by reason of the coning of the wheels, the variations in the movement of the locomotive, and by irregularities in track conditions. Variations in the design of locomotives, such as the length of wheel base, the number of drivers, the use of two-wheel or four-wheel trucks may be made without bringing unduly high stresses into the rail, provided proper care is used in making the design. On curved track other considerations will be found to enter into the development of stresses in the track. The change of direction of the locomotive in going around a curve is effected by pressure between the flanges of the outer wheels of the front truck and the outer rail and frequently also by the outer first driver, resulting in a lateral bending of the outer rail. An unexpected effect is the greatly increased load transmitted to the inner rail by one or more of the drivers at low and medium speeds and a corresponding decrease in the load on the opposite driver, the conditions producing excessive vertical bending stresses in the inner rail. The lateral spreading action on the inner rail develops marked lateral bending stresses in that rail. In all of these matters the characteristics of the type of locomotive will be found to influence the action of the locomotive and the nature and amount of the stresses developed in the rail under the various wheels.

In the matter of changing the direction of the locomotive it seems evident from the tests that the four-wheel truck of the Pacific and Mountain types presents advantages over the two-wheel truck of the Santa Fe and Mikado types. With the four-wheel leading truck used, the loads on the wheels are such as to give moderate vertical bending stresses. The lateral bending stresses in the outer rail under the outer wheels of the four-wheel truck on the curves from 4° to 10° may be termed moderate. With the locomotives having the four-wheel trucks there is little or no outward bending stress developed in the outer rail under the first driver, the first driver participating in the turning action only at the highest speeds. With the two-wheel front truck the lateral bendinng stress in the outer rail under the first driver may become very great, and the value 22,600 lb. per sq. in. found under the first driver of the Santa Fe type on the 10° curve at a speed of 35 miles per hour may be considered a very excessive stress. It would appear that the use of a four-wheel truck would improve the conditions under the front outer driver of this locomotive. In the case of the Double Trailer Santa Fe type the load has evidently been redistributed in such way as to give an increased vertical load upon the wheels of the front truck. (It will be recalled that this locomotive was not redesigned for the use of the four-wheel trailer.) The resulting lateral bending stress at the outer wheel of the truck is very great, 16,000 lb. per sq. in.

The lateral bending stress in the inner rail under the intermediate drivers of these locomotives is so important that its presence and possible magnitude are matters that should receive careful consideration in designing such locomotives. It is evident that the magnitude of the

lateral bending stress in the inner rail may be expected to depend upon the wheel base, though the lateral bending stress under the third driver of the Mountain type on the 10° curve is nearly as great as that under the fourth driver of the Santa Fe type, 19,800 as compared with 23,500 lb. per sq. in. The driver wheel base of the former is 18 ft.; of the latter, 22 ft. The connection of the front truck of the Mountain type (four-wheel) may be less flexible than that of the Santa Fe type locomotive (two-wheel).

It is of interest to note the effect of removing flanges from intermediate drivers. The Santa Fe type locomotive of the A. T. & S. F. Ry. ordinarily has only the third drivers flangeless. In the tests at Cajon both the third and the fourth drivers of the locomotive were equipped with flangeless tires (blind drivers). This ommission of the flange and the use of the cylindrical tire shifted the very high vertical bending stress in the inner rail from the fourth driver to the fifth driver, and the vertical bending stress was changed from 29,000 lb. per sq. in. under the fourth driver to 29,000 lb. per sq. in. under the fifth driver. The highest lateral bending stress was also shifted to the fifth driver and changed to 25,900 lb. per sq. in. The Heavy Santa Fe type locomotive of the S. P. Co. used in the tests on the 10° curve at Bealville was equipped with flanges on all the drivers, the regular practice of this railroad. In these tests the greatest lateral bending stress was under the fourth driver, but the third driver helped considerably, and the lateral bending stress under the fourth driver was smaller than that found with the locomotive of the A. T. & S. F. Ry. on the same 10° curve; 18,600 for the former as compared with 25,800 lb. per sq. in. for the latter.

It should be noted that the axle of the first driver of the locomotive of the S. P. Co. was designed to have some lateral play in the bearings. The lateral bending stress in the outer rail under the first driver at the speeds used differed so little in nature and magnitude from those found with the locomotive of the A. T. & S. F. Ry. that it appears that this play had no effect upon the guiding action of this driver or upon the lateral bending stresses developed by the intermediate drivers. It seems also that the outward bending or deflection of the inner rail and consequent temporary increase of gage at the fourth and fifth drivers in case the third driver is flangeless, and also at the third driver in case it is flanged, throw light upon the ability of a rigid wheel base to pass around a sharp curve without permanently spreading the track, even though the flange of the outer rear driver keeps away from the outer rail.

The lateral bending stresses under the trailers of the Mountain and Santa Fe types have high values on the 10° curve though they are moderate on the 6° curve. It is evident that there is sufficient freedom in turning to give low lateral bending stresses on the 6° curve, but that the restraint on the 10° curve develops high lateral pressures against the outer rail. The double trailer Santa Fe type, however, has such freedom of motion that relatively small lateral bending stresses are

developed under these wheels on the 10° curve. It should also be recorded that the lateral springs regulating the lateral swing of the trailer truck in the Mountain type locomotive were loosened for one day's runs; the lateral bending stresses developed in the two rails of the 10° curve with the trailer truck in this condition did not differ particularly from those found when the springs were in their usual condition.

22. **Effect of Pulling and Coasting and of Condition of Rail.**— In the discussion of the tests on straight track it has been shown that the stress in rail under the wheels of the locomotive does not differ greatly whether the locomotive is pulling a load or coasting. On the curved track at Bealville, California, the average of the vertical bending stresses in base of rail for both outer and inner rails of the 10° curve at all the wheels of the locomotive did not differ noticeably whether the locomotive was pulling a load of 500 tons up-grade or whether it was coasting down-grade. This load was about half the full capacity of the locomotive in operating on the 2.2 per cent grade. The stresses under individual wheels, however, showed some differences.

At Cajon the stress in the outside edge of the inner rail under the fifth driver (see Fig. 91 to 94) was somewhat greater when the locomotive was pulling load than when coasting for those tests when the tire of the fourth driver was cylindrical (typical of new blind tire) and only slightly greater in the tests when the fourth driver was grooved. It is seen then that the lateral bending stresses were about the same in the two cases. The vertical bending stresses in the outer rail under the fifth driver were somewhat smaller when the locomotive was pulling. The stresses in the inner rail under the second and third drivers were somewhat smaller when the locomotive was pulling a load than when coasting. The difference in the stresses in the outer rail at the first driver and at the wheel of the front truck (and also the lateral bending stress at other wheels) do not indicate a change that can be considered characteristic. Altogether no very marked differences in stress under the various wheels are noticeable in the two ways of operating the locomotive.

The comparison of the results of the tests at Bealville is complicated by the fact that the down-grade tests were made on wet rails, while in the up-grade tests the rails were dry. It is seen, however, that the vertical bending stress under the rear driver, which is the location giving apparently somewhat higher stresses when the locomotive is pulling load and the one for which higher stresses may possibly be expected for this condition if at all, does not differ in either rail for the up-grade and down-grade tests with either locomotive. It is apparent, however, that there is a considerable difference in the distribution of the stresses under the intermediate drivers due no doubt to differences in friction of wheel on rail for the two conditions of wet and dry rail and, therefore, to differences in the position of the center of rotation in the change of direction of the locomotive on the curve. At the outside edge of the

inner rail under the fourth driver of the locomotive of the A. T. & S. F. Ry. the stress drops from 54,700 lb. per sq. in. when the rail is dry (up-grade, pulling) to 36,600 lb. per sq. in. when the rail is wet (down-grade, light). The stresses under the first, second and third drivers are somewhat less with the dry rail. The lateral bending stresses in the outer rail under the first driver and the wheel of the front truck are higher with the wet rail. With the locomotive of the S. P. Co. a reduction in stress with the wet rail is found at the main driver, that at the fourth driver remaining the same. It is evident that the condition of the rail has an effect upon the development of stress at the different wheels.

23. **Tests With Locomotive Backing Over Curved Track.**—On curved track with the locomotive moving forward the guiding and the change of direction around the curve are effected by the flanges of the outer wheel of the front truck and generally also by that of the first driver. When the direction of the locomotive is reversed it is to be expected that the turning action will be effected by the fifth driver and also by the trailer if its connections are not too flexible. It is also to be expected that the inner second driver will cause large lateral and vertical bending stresses in the rail at the point where the fourth driver develops these large stresses when the locomotive is moving forward. That this reasoning is correct is shown by an examination of Fig. 99 and 100, which give stresses in rail when the Santa Fe type locomotives of the two railroads were backed around the 10° curve at 10 miles per hour, the rails being dry in these tests. The magnitude of the stress at these wheels is not as great as may be expected. It will be noted that with the locomotive of the A. T. & S. F. Ry. the turning action is effected by the pressure of the flange of the trailer against the outer rail, while with the locomotive of the S. P. Co. the turning action is taken more largely by the fifth driver. Evidently the trailer of the former locomotive is more restrained in its connections. This restraint was also noticed with a locomotive of the same type in the tests on the 10° curve at Ribera (the locomotive moving forward), but as that restraint did not show up on the 6° curve at Ribera it would seem that there is sufficient freedom of action in the connection of the trailer not to develop lateral bending stresses under the trailer on the 6° curve.

24. **Lateral Bending of the Rail Section.**—When a lateral force is applied to the head of the rail and the base of the rail is held in place either wholly or partly, it may be expected that the rail web will bend and the head of the rail will deflect outwardly or inwardly relatively to the original vertical axis of the rail. To determine whether this lateral bending of the rail section is a measurable quantity, an instrument was made which was fastened firmly to the base of the rail and had a vertical standard rising at one side of the rail. An Ames dial at the level of the middle of the depth of the head of the rail was attached to this standard; its plunger bore against the side of the head of the rail. The locomotive

was run by at a slow speed, 1 to 2 miles per hour, and readings were taken as a wheel passed the instrument and again between wheels. The observation proved to be definite and positive and succeeding runs confirmed the readings. It should be noted that the observations are independent of any tilting of the rail about its base or its support upon the tie; such tilting may also occur.

Measurable values of the lateral bending of the rail section were found. Reference will be made only to the general nature of the results. The deflection varied from 0 to 0.016 in. The values on the 10° curve were somewhat higher than on the 6° curve. The Santa Fe type gave somewhat higher values than the Mountain type. As would be expected from the low speed used the deflections were greatest in the inner rail and the direction of deflection in the inner rail was generally away from the track. The direction of the deflection in the outer rail did not always correspond with the character of the observed lateral stress in the base of rail. Since the lateral bending of the rail itself is the result of a number of forces including those acting at the supports of the rail, it is possible that the sign of the bending moment at a wheel point may not be in keeping with the direction of the lateral force acting at that point. An interesting feature occurred when the Santa Fe type was backed over the track. A very high value of the deflection was observed in the inner rail under the fifth driver, a deflection of 0.016 in. away from the track for the 10° curve and 0.014 in. for the 6° curve. The absence of leading wheels to guide the driving wheel base is evident in these results. Values of the deflection as great as 0.007 in. were found at the wheels of the tender, the middle wheel of a truck on the inner rail giving large deflections.

25. **Preliminary Tests on the Illinois Central Railroad.**—Preliminary to the main tests on curved track heretofore described, tests were made on two curves on the Illinois Central Railroad for the purpose of developing the method of testing curved track, of learning the technique necessary for a successful test, and of trying out the instrumental equipment. The results are of sufficient interest to warrant recording.

Fig. 101 gives the stresses in the inner and outer rail of a 7° curve at Murphysboro, Illinois, with a Mikado type of locomotive (No. 1861) of the same general class as that used in the tests on straight track described in the second progress report, the rail being 90-lb. A.R.A.-A. section, while in the earlier tests on straight track the rail was 85-lb. A.S.C.E. section. The superelevation of the track was 3¼ in., corresponding to a speed of 28 miles per hour. It is seen that the turning action is taken by the flanges of the outer front truck wheel and the first driver, and there is little difference in lateral bending stresses under these wheels at the two speeds used. The fourth driver and the trailer push outwardly on the outer rail at both speeds; the trailer also acts to increase the curvature of the inner rail. The main lateral bending stresses in the

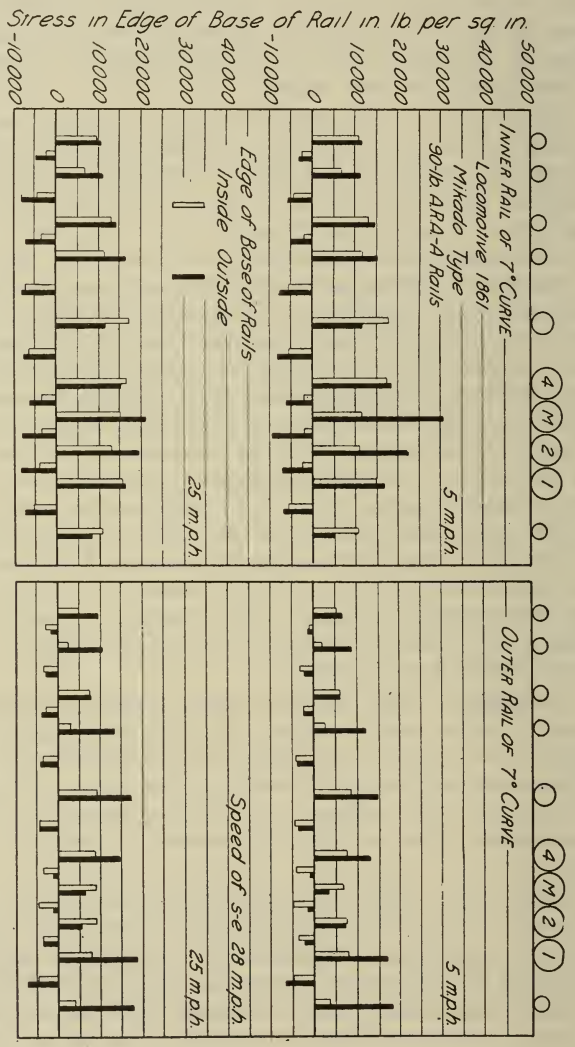


Fig. 101.—Stress at Inside and Outside Edges of the Base of the Inner and Outer Rails of the 7° Curve—Series 5056-62, Mikado Type Locomotive of the I. C. R. R.

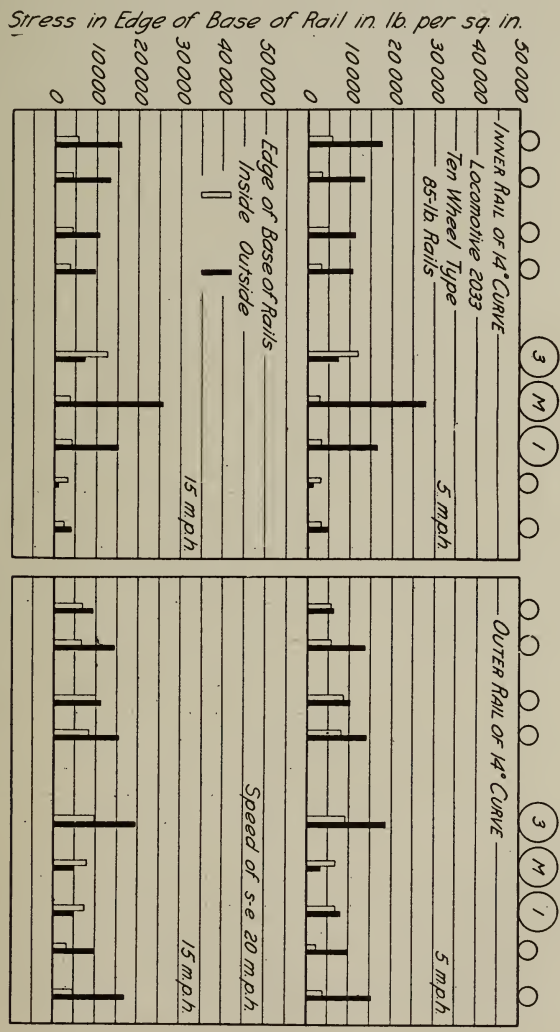


Fig. 102.—Stress at Inside and Outside Edges of the Base of the Inner and Outer Rails of the 14° Curve—Series 5035-41, Ten-wheel Type Locomotive of the I. C. R. R.

inner rail are at the second and third drivers. The vertical bending stress at the third driver at a speed of 5 miles per hour is 80 per cent more than that on straight track and at 25 miles per hour is 30 per cent greater than on straight track. In the outer rail the stresses at the second and third drivers are very small.

Fig. 102 gives the stresses in the inner and outer rail of a 14° curve connecting the Havana Branch at Champaign, Illinois. The locomotive is a ten-wheel type (4-6-0). The wheel loads are relatively small. The rail is 85-lb. section. The superelevation of the track, 3¾ in., corresponds to a speed of 20 miles per hour. It is seen from the stresses that the turning of the locomotive at both speeds is produced by the action of the flanges of the two outer wheels of the front truck. The third driver pushes outwardly on the outer rail at both speeds. The first and second drivers push outwardly on the inner rail, the lateral bending stress under the second driver being of considerable magnitude at both speeds. The vertical bending stress in the inner rail at the second driver is nearly 50 per cent greater than on straight track for both speeds. It is seen that even this type of locomotive shows great variation in stress on this curve.

26. **Progress of Other Work.**—Other investigations not reported here have been carried on. Some further progress has been made on the transmission of pressure through granular materals. A few tests made with a loaded car equipped with a wheel having a large flat spot showed that strains of considerable magnitude are caused by this condition and proved the practicability of measuring these strains by means of the stremmatograph.

Laboratory tests on several of the forms of rail joints commonly used in track have been conducted. In these tests measurements of the strains in different parts of the bars and the rail under ordinary loads and somewhat higher loads have been made with a view of learning where stresses are developed and how they are transmitted between rail and joint, in an effort to learn more of the characteristics of the joints. A few tests to find what stresses are developed in the joints in track have also been made.

In the test program under consideration are included the effect of canted rails, stresses under heavily loaded cars, and tests with electric locomotives where the conditions of center of gravity, relation of unsprung to sprung weight, and spacing of wheels may be quite different from those of the steam locomotive.

Respectfully submitted,
THE SPECIAL COMMITTEE ON
STRESSES IN RAILROAD TRACK,
By A. N. TALBOT, *Chairman.*

November 29, 1922.

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